Shelter Research 1400
Component Development
Subtask 1411C
Large Auxiliary
Power Systems

SUMMARY

OF

RESEARCH REPORT

MINIMUM REQUIREMENTS FOR AUXILIARY POWER SYSTEMS FOR COMMUNITY SHELTERS

July 30, 1964

This is a summary of a report which has been reviewed in the Office of Civil Defense and approved for publication. Approval does not signify that the contents necessarily reflect the views and policies of the Office of Civil Defense.

OCD-OS-62-190
Subtask 1411C

BATTELLE MEMORIAL INSTITUTE
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Columbus, Ohio 43201

Summary Prepared
By Battelle Memorial Institute
July 1964

SUMMARY

of

RESEARCH REPORT

on

MINIMUM REQUIREMENTS FOR AUXILIARY POWER SYSTEMS FOR COMMUNITY SHELTERS

SCOPE

The scope of the study was defined by the following assumptions:

(1) Power systems were to have a minimum output equivalent to 5 kw and no limited maximum output except that imposed by the availability of conventional commercial components. Power systems smaller than 5 kw and also all of those that use unconventional devices such as fuel cells and thermoelectric modules are the subjects of separate studies conducted by other OCD contractors.

(2) A period of 10 years was selected as a reasonable time in which the shelter might be expected to be on "stand-by". It was assumed that period would be followed by a two-week occupancy period. During the first 24 hours of occupancy, the shelter might have to be completely isolated from the outside atmosphere. These operational periods represent a reasonable estimate of what might actually occur; however, their selection is not intended to imply that these are the most representative periods which could be selected.

(3) To make the results of the study as broadly applicable as possible, it was assumed that the shelter could be located anywhere in the United States.

OBJECTIVES

Because of the variety of power-system components available to a shelter designer and because of the nonstandard operating conditions, it is necessary for OCD to determine the effects of these nonstandard conditions on component performance and to develop the criteria and parameters for the design, installation, operation, and maintenance of auxiliary power systems including supporting equipment.

The objectives of this research program were, therefore:

(1) To determine the most appropriate types of power systems for various shelter systems

(2) To define the requirements of design, installation, and maintenance of the power system and its auxiliaries

(3) To demonstrate the operating capabilities of a power unit under representative environmental conditions.

APPROACH

The research program was divided into four general categories:

(1) Evaluation of prime movers

(2) Evaluation of power-utilization systems

(3) Development of requirements for installation, operation, and maintenance of complete systems including supporting equipment

(4) Demonstration of the validity of the recommended specifications by designing and assembling a 20-kw unit and operating it in an underground structure.

In conducting the research, literature studies were made in the various areas of interest, and design and performance data on manufacturers' equipment were studied. Numerous persons with experience pertinent to the study were consulted. Components and systems believed most suitable for use in community shelters were analyzed and compared. Finally, an experimental investigation was carried out in the Battelle underground testing facility.

RESULTS AND CONCLUSIONS

The results of this study show that the minimum requirements for installation, operation, and maintenance of shelter auxiliary power systems can generally be met with commercially available equipment now in common industrial use. However, a number of technological areas were revealed in which the presently available equipment and experience are limited in meeting the specific needs of community shelters. These areas are:

(1) Sealed-period operation of a prime mover

(2) Manually operated starting systems requiring no maintenance

(3) Low-cost preservation and storage techniques for fuels and equipment requiring minimum reactivation

(4) Low-cost waste-heat recovery equipment.

Because of the importance of these subjects in the over-all community shelter program, Battelle recommends that further research be conducted to determine feasible and practical solutions to the unsolved problems.

SYNOPSIS

The Office of Civil Defense initiated and supported this research program as part of its effort to develop designs of community-size protective shelters that have maximum cost effectiveness. The program had as its over-all objective the determination of the minimum requirements for auxiliary power systems that might be installed in various types of shelters anywhere in the United States. The criteria developed in the study are to be used in the preparation of manuals for shelter construction.

Community-size shelters, designed to accommodate large numbers of occupants, must be provided with power for essential services such as cooking, communications, lighting, pumping, and environmental control. To ensure availability of power during an emergency, it will be necessary to include a self-contained auxiliary power system as part of the permanent shelter equipment since commercial power is likely to be disrupted.

The shelter designer has a wide range of commercial equipment from which to select the system components which would be most suitable for a specific installation. Commercially available prime movers include compression-ignition and spark-ignition engines, gas turbines, and steam generator-steam engine or turbine combinations. Any of these prime movers can be used to drive electric generators, air compressors, or hydraulic pumps, which in turn can power the various shelter subsystems. Major items of shelter equipment such as blowers and cooling systems can also be driven directly by the prime mover with the addition of a small electric generator to supply power for lights and for other low-power equipment dependent exclusively on electricity.

Shelter power systems can be assembled largely from conventional commercially available components. However, because of the circumstances under which shelter power systems can be required to operate, it is necessary to give special consideration to such factors as fuel storage, waste heat removal and utilization, safety, reliability, maintenance, and noise and vibration.

The report is divided into 10 major sections, namely:

(1) Prime movers

(2) Starting systems

(3) Cooling

(4) Fuel storage

(5) Waste heat recovery

(6) Power transmission systems

(7) Mountings and drives

(8) Noise and vibration

(9) Stand-by maintenance

(10) Demonstration unit.

Prime Movers

The prime-mover types studied were: compression-ignition engines, spark-ignition engines, gas turbines, and steam power systems. Each type was evaluated on the basis of: physical characteristics; performance, control, installation and maintenance requirements; safety and human comfort; and economics.

The steam power system was judged impractical because of: high first cost; high operating cost; high maintenance cost, extra space requirements; and complexity of equipment, operation, and control.

Starting Systems

Electric, hydraulic, pneumatic, and manual starting systems were studied. Each was evaluated on the basis of physical characteristics, maintenance, dependability, and economics. The manual starting systems investigated were of the energy-storage type and included: flywheel, falling weight, and spring-types. No commercial versions of these starting systems are presently available; however, these systems show potential for being less costly, easier to maintain for long periods, and significantly more reliable than the commercially available starting systems.

The use of starting aids such as engine-block and oil-sump heaters and starting fluids for the intake air were also considered. These starting aids would be useful under most conditions to improve the dependability of starting.

Cooling

Direct make-up-water, heat-exchanger, radiator, and ebullient cooling systems were studied. Each was evaluated on the basis of physical characteristics, installation, maintenance, dependability, water or air consumption, and economics.

Ventilation of the engine room to remove heat lost from the engine and other components by radiation will be essential. These losses may be up to half as much as the jacket-water cooling losses. Ventilating the engine room with exhaust air from the occupied section of the shelter may be possible and would be desirable in that a minimum of additional equipment would be required.

Gas turbines require no jacket-water cooling system and are likely to radiate no more heat to the engine room ventilating air than piston engines.

Fuel Storage

The storage capabilities of the following fuel types were evaluated: 90-octane gasoline, straight-run kerosine, No. 2 diesel oil, and liquefied petroleum gas (LPG).

All fuels deteriorate in storage to some degree, dependent on the type of fuel, the storage conditions, and the length of storage time. The primary causes of deterioration in fuels are: evaporation, oxidation, contamination, and polymerization. Any fuel-storage technique which inhibits any of these will increase the storage life of the fuel. Fuel deterioration can be detrimental in reducing ignition and burning qualities of the fuel, increasing corrosion in the fuel system, increasing clogging and fouling in the fuel system, and increasing contamination of the lubricating oil.

Two basically different fuel-storage techniques may be followed: active storage and long-term storage. In an active fuel-storage program a fuel would be replaced or replenished at regular intervals, and the storage-system requirements would be relatively uncritical. In a long-term fuel-storage program, the storage system would be designed to preserve the fuel for the longest possible period of time. For an active fuel-storage system a simple vented tank, either above-ground or (preferably) underground, would be suitable. A sealed underground tank would significantly increase the potential storage life of most fuels and would be suitable for a long-term fuel-storage system. Adding a pressurized nitrogen "blanket" to the sealed storage tank would further increase storage life.

Waste Heat Recovery

All conventional prime movers convert only part (up to about 1/3) of the total fuel energy supplied to useful shaft work, the remainder being rejected as waste heat to the cooling and exhaust systems or lost as radiation. For piston engines the amounts of waste heat in the coolant and in the exhaust gases are each approximately equivalent to the shaft power output. For regenerative gas turbines the exhaust waste heat is approximately 2-1/2 times the shaft power output.

The waste heat from the cooling system can be fairly easily recovered either: (1) as hot water from a water-to-water heat exchanger or from the jacket water directly, or (2) as low-pressure steam and/or hot water directly from the steam separator tank of an ebullient cooling system. Only about 60 to 80 per cent of the exhaust waste heat is recoverable because of the danger of corrosion in the exhaust system if the gases are cooled below 300 F. Waste heat recovered from the exhaust system can be made available as hot water or as high- or low-pressure steam.

Recovery of waste heat for use in providing a hot-water supply or for space heating is practical wher there is a reasonable demand for these services in the shelter. It is more advantageous to recover waste heat from the cooling system because: (1) less extra equipment is needed, (2) the exhaust waste heat

is easily rejected from the shelter, and (3) there is a danger of overheating and corrosion with an exhaust heat-recovery system that does not exist with a cooling water heat-recovery system.

Recovery of waste heat for shaft power is not practical because of the cost and complexity of the extra equipment required.

Power Transmission Systems

Mechanical, electric, hydraulic, and pneumatic power-transmission systems were studied. Each was evaluated on the basis of physical characteristics, over-all efficiency, installation and maintenance requirements, reliability, safety and human comfort, and economics. The mechanical, hydraulic, and pneumatic systems all require a small electric generator to supply the lighting and communications requirements of the shelter. The electric power-transmission system has an advantage in that it can be more easily integrated into a commercial power system for independent exercising of individual components, for potential use of the shelter space during the stand-by period, and for use of commercial power if it is available in an emergency.

Mountings and Drives

Five items to be considered in designing or selecting mountings and drives are: (1) alignment, (2) vibration, (3) piping connections, (4) load characteristics, and (5) blast effects. Mounting and drive components should be selected to prevent annoying or destructive vibration from developing within the mounted components and from being transmitted to or from other parts of the shelter.

Applicable mounting techniques are foundation mounting or skid mounting, and flexible or rigid mounting. Flexible skid mounting is preferable for community shelter auxiliary power systems as it would result in acceptable component alignment, minimum vibration transmission and low installation cost.

Of the applicable drive techniques, a flexible direct-coupling drive is preferable for community-shelter auxiliary-power systems as it would be compatible with skid mounting and would function with a minimum of vibration transmission. A clutch is necessary when the driven-component starting torque is high or when the prime mover and driven component shafts must be rotated independently.

Noise and Vibration

Reliable and useful data on the noise generation characteristics of prime movers are not presently available. However, from general experience and limited test measurements made at actual installations and in the Battelle test facility it is apparent that some noise control measures will be required with most prime movers. However, the noise problem should not be difficult to solve.

The chief sources of noise from prime movers are vibrating surfaces and aerodynamic pulsations. Control of this noise can be effected in the following ways:

(1) Careful selection of the prime mover and its auxiliary equipment

(2) Provision of barriers between the prime mover and the shelter occupied space

(3) Reduction or elimination of transmission of vibration from the source

(4) Use of sound absorbing materials on the walls and ceiling of the occupied shelter space

(5) Providing "sound trap" air ducts when noise source area and occupied space must be connected for ventilation.

The experimental work in the Battelle test facility indicates that thick walls, complete closure, and sound-trap ducting would be effective in reducing the noise reaching the occupied space to an acceptable level. It was also noted that no special vibration isolation equipment would be necessary to prevent annoying engine vibration from reaching the occupied space. The normal construction of underground shelters would be adequate to dampen the vibration.

Stand-by Maintenance

Two general approaches to stand-by maintenance were considered in this study: dynamic maintenance and static maintenance. Dynamic maintenance involves frequent and periodic exercising and inspection and is employed almost universally for conventional emergency stand-by power systems. Static maintenance involves long-term storage with infrequent exercising and inspection.

Dynamic maintenance is essential if instant starting and load assumption is required in the event of commercial power failure. The procedures for dynamic maintenance are fairly well established from experience. A minimum frequency of once every six weeks and a 1 to 2 hour full-load run are advisable. Periodic changes of lubricants and coolant are necessary to prevent damage or inoperability due to time-dependent deterioration of these liquids.

Static maintenance may necessitate allowing a period of 1 or 2 hours for reactivation of the equipment. During reactivation a community shelter could be occupied with emergency lighting and communications supplied by long-shelf-life batteries. A satisfactory static maintenance technique could reduce the long-term maintenance costs of a community shelter program a significant amount. The static maintenance program would require maintaining a low (30 to 35 percent) humidity level in the shelter, keeping liquids that tend to deteriorate away from components which may be easily fouled or otherwise rendered inoperable, and coating critical surfaces with a long-life preservative.

Static maintenance techniques presently in use, such as "mothballing" of surplus military equipment, are not directly applicable to the community shelter because of cost and the time required (a minimum of several days) to reactivate. However, the available experience and proven techniques are sufficiently encouraging to indicate the potential feasibility of the approach.

Demonstration Unit

A 20-kw diesel engine-generator set was set up in an underground test facility to experimentally verify the parameters developed in this study in a simulated community shelter environment. The same unit was also used to evaluate the performance of simple exhaust heat-recovery systems.

As part of the experimental studies, two test runs were made to determine the system heat balance and to measure the effect of insulating the exhaust system. For one run the exhaust system was not insulated, for the other run 1-in.-thick insulation was applied to the exhaust manifold and tubing in the engine room. The test results effectively demonstrated that insulating the exhaust system reduced the amount of heat rejected to the cooling system, ventilating air, and walls, thus reducing the shelter heat sink requirements.

The results of the demonstration program verified the validity of the parameters developed in the study.

Because commercial exhaust heat-recovery equipment is costly, several simple configurations were studied to determine their effectiveness. About 80 per cent of the heat in the exhaust gases is recoverable without encountering exhaust system corrosion. A straight 3/4-in.-diameter brass tube installed in the exhaust system had a recovery efficiency of 17 per cent. A coiled tube and a straight-finned tube both 3 ft long, had recovery efficiencies of 36 per cent. From these studies it can be concluded that simple, inexpensive exhaust heat-recovery systems are effective and practical.

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July 29, 1964

Office of Civil Defense
Department of Defense
The Pentagon
Washington 25, D.C.

Attention Director for Research

Dear Sir:

Attached is our Summary Report, "Minimum Requirements for Auxiliary Power Systems for Community Shelters", prepared under Contract No. OCD-OS-62-190, Subtask 1411C. We are distributing copies of this report in accordance with the Shelter Research Program Standard Distribution List, including Attachment 1, which was forwarded to us by the Deputy Assistant Director for Research.

A principal result of this study is the conclusion that the minimum requirements for installation, operation, and maintenance of shelter auxiliary power systems can be met with commercially available equipment now in common industrial use. This represents a significant advantage since both low-cost equipment and operating experience will be available to the shelter designer.

However, this study also pinpointed a number of technological areas in which development of new equipment and techniques would reduce the cost of setup and maintenance, improve the reliability, and broaden the capabilities of shelter auxiliary power systems. Battelle strongly recommends further research on shelter auxiliary power systems with the following basic objectives: (1) prime movers capable of operating at full or part load without an external air supply for up to 24 hours, (2) low-cost manually operated stored-energy type starting systems requiring no maintenance, (3) low-cost preservation and storage techniques for fuels and equipment requiring minimum attention and reactivation time and effort, and (4) low-cost waste-heat recovery equipment specifically tailored to the shelter requirements.

In recognition of the need for further research Battelle has submitted two proposals to the Office of Civil Defense. These are: "Development of Closed-Cycle Combustion System for Shelter Auxiliary Power Plant", submitted February 20, 1964, and "Development of Manually Operated Stored-Energy Starting System for Shelter Auxiliary Power Plant",

submitted April 14, 1964. Copies of both of these proposals were also sent to the Civil Defense Technical Office at Stanford Research Institute on June 26, 1964.

We have appreciated the opportunity to conduct this research program for the Office of Civil Defense. We shall be glad to receive questions or comments concerning this report or the suggested future research.

Sincerely,

James A. Eibling

James A. Eibling
Group Director
Thermal Systems Research

JAE:mvv

Enc.

FOREWORD

The Office of Civil Defense initiated and supported this research program as part of its effort to develop designs of community-size protective shelters that have maximum cost effectiveness. The program had as its objective the determination of the minimum requirements for auxiliary power systems that might be installed in various types of shelters anywhere in the United States. The criteria developed in the study are to be used in the preparation of manuals for shelter construction. Consequently, the report is written primarily for technically trained users who have general knowledge of power systems.

This program was carried out at Battelle Memorial Institute by personnel in the Thermal Systems Group of the Mechanical Engineering Department. Mr. Frank C. Allen of the Office of Civil Defense Research Directorate staff monitored the work and contributed to the over-all planning of the research. His assistance and guidance is greatly appreciated. The authors also wish to acknowledge the contributions of other Battelle staff members, particularly W. L. Buckel, R. D. Wilson, B. Underwood, and R. D. Fannon.

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MINIMUM REQUIREMENTS FOR AUXILIARY POWER SYSTEMS FOR COMMUNITY SHELTERS

by

D. A. Trayser, L. J. Flanigan, and S. G. Talbert

INTRODUCTION

Community-size shelters, designed to accommodate large numbers of occupants, must be provided with power for essential services such as cooking, communications, lighting, pumping, and environmental control. Commercial power might be disrupted in an emergency of the severity that would require occupancy of the shelter. To ensure availability of power during such an emergency, it will be necessary to include a self-contained auxiliary power system as part of the permanent shelter equipment.

Shelter power systems can be assembled largely from conventional commercially available components. However, because of the circumstances under which shelter power systems will be required to operate, it is necessary to give special consideration to such factors as fuel storage, waste heat removal and utilization, safety, reliability, maintenance, and noise and vibration.

The shelter designer has a wide range of commercial equipment from which to select the system components which would be most suitable for a specific installation. Commercially available prime movers include compression-ignition and spark-ignition engines, gas turbines, and steam generator-steam engine or turbine combinations. Any of these prime movers can be used to drive electric generators, air compressors, or hydraulic pumps, which in turn can power the various shelter subsystems. Major items of shelter equipment such as blowers and cooling systems can also be driven directly by the prime mover with, for instance, a small electric generator to supply power for lights and for other low-power equipment dependent exclusively on electricity.

Because of the variety of power-system components available to a shelter designer and because of the nonstandard operating conditions, it is necessary for OCD to determine the effects of these nonstandard conditions on component performance and to develop the criteria and parameters for the design, installation, operation, and maintenance of auxiliary power systems including supporting equipment.

The scope of the study was defined by the following assumptions:

(1) Power systems were to have a minimum output equivalent to 5 kw and no limited maximum output except that imposed by the availability of conventional commercial components. Power systems smaller than 5 kw and also all of those that use unconventional devices such as fuel cells and thermoelectric modules are the subjects of separate studies conducted by other OCD contractors.

(2) A period of 10 years was selected as a reasonable time in which the shelter might be expected to be on "stand-by". It was assumed this period would be followed by a two-week occupancy period. During the first 24 hours of occupancy, the shelter might have to be completely isolated from the outside atmosphere. These operational periods represent a reasonable estimate of what might actually occur; however, their selection is not intended to imply that these are the most representative periods which could be selected.

(3) To make the results of the study as broadly applicable as possible, it was assumed that the shelter could be located anywhere in the United States.

During this study it was found that in several areas, presently available equipment and information do not adequately fulfill the needs for the design of shelter auxiliary power systems. These needs are discussed in the "Future Research Needs" section of this report.

In conducting the study literature studies were made in the various areas of interest and design and performance data on manufacturers' equipment were studied. Numerous persons with experience pertinent to the study were consulted. Components and systems believed most suitable for use in community shelters were analyzed and compared. Finally, an experimental investigation was carried out in the Battelle underground testing facility.

SUMMARY

The results of this study show that the minimum requirements for installation, operation, and maintenance of shelter auxiliary power systems can generally be met with commercially available equipment now in common industrial use.

To facilitate the use of the information in this report, the presentation of results is divided into ten sections. The material in these sections is summarized here in the same sequence, namely:

(1) Prime movers
(2) Starting systems
(3) Cooling
(4) Fuel storage
(5) Waste heat recovery
(6) Power transmission systems
(7) Mountings and drives
(8) Noise and vibration
(9) Stand-by maintenance
(10) Demonstration unit.

Information and data were obtained from the literature on equipment and systems and on the present state of the art. In addition, information was obtained from manufacturers and knowledgeable individuals in the various fields covered by the study.

Prime Movers

Compression-ignition engines, spark-ignition engines, gas turbines, and steam power systems were included in this study area. Each prime-mover type was evaluated on the basis of: physical characteristics; performance, control, installation, and maintenance requirements; safety and human comfort; and economics.

Table 1 and Figures 1 through 5 summarize the results of these evaluations. Table 1 shows general qualitative comparisons between the prime movers which were judged the most applicable for use in community shelters. Figures 1 and 2 show specific volume and weight data for these same prime movers. Figures 3 and 4 show specific fuel consumption and combustion air requirements, and Figure 5 shows approximate purchase prices.

The steam power system was omitted from these comparisons as it was judged impractical because of: high first cost; high operating cost; high maintenance cost; extra space requirements; and complexity of equipment, operation, and control.

TABLE 1. COMPARISON OF PRIME MOVERS

Evaluation Parameter	Prime Mover: Diesel Engine	Gasoline Engine	LPG Engine	Gas Turbine	Steam Engine
Installation	Simple	Simple	Simple	Very Simple	Complex
Maintenance	Low	Moderate	Moderate	Very Low	High
Dependability	Good	Fair	Good	Good	Good
Safety	Good	Fair	Fair	Good	Fair
Availability	Excellent	Good	Good	Poor	Fair
Cost	Presented in Fig. 5				

Starting Systems

Electric, hydraulic, pneumatic, and manual starting systems were included in this study area. Each was evaluated on the basis of physical characteristics, maintenance, dependability, and economics.

Table 2 summarizes the results of these evaluations. Several manual, energy-storage starting techniques were investigated briefly during this study. These were: flywheel type, falling weight type, and spring type. No commercial versions of these energy-storage starting systems are presently available, however, these systems show potential for being less costly, easier to maintain for long periods, and significantly more reliable than the commercially available starting systems.

The use of starting aids such as engine-block and oil-sump heaters and starting fluids for the intake air were also considered. These starting aids would be useful under most conditions to improve the dependability of starting.

TABLE 2. COMPARISON OF STARTING SYSTEMS

Starting System	Maintenance	Dependability	Availability	Cost
Electric	High	Fair	Low	Good
Hydraulic	Low	Good	High	Good
Pneumatic	Low	Fair	High	Good
Manual energy-storage	Negligible	Excellent	Low	None available

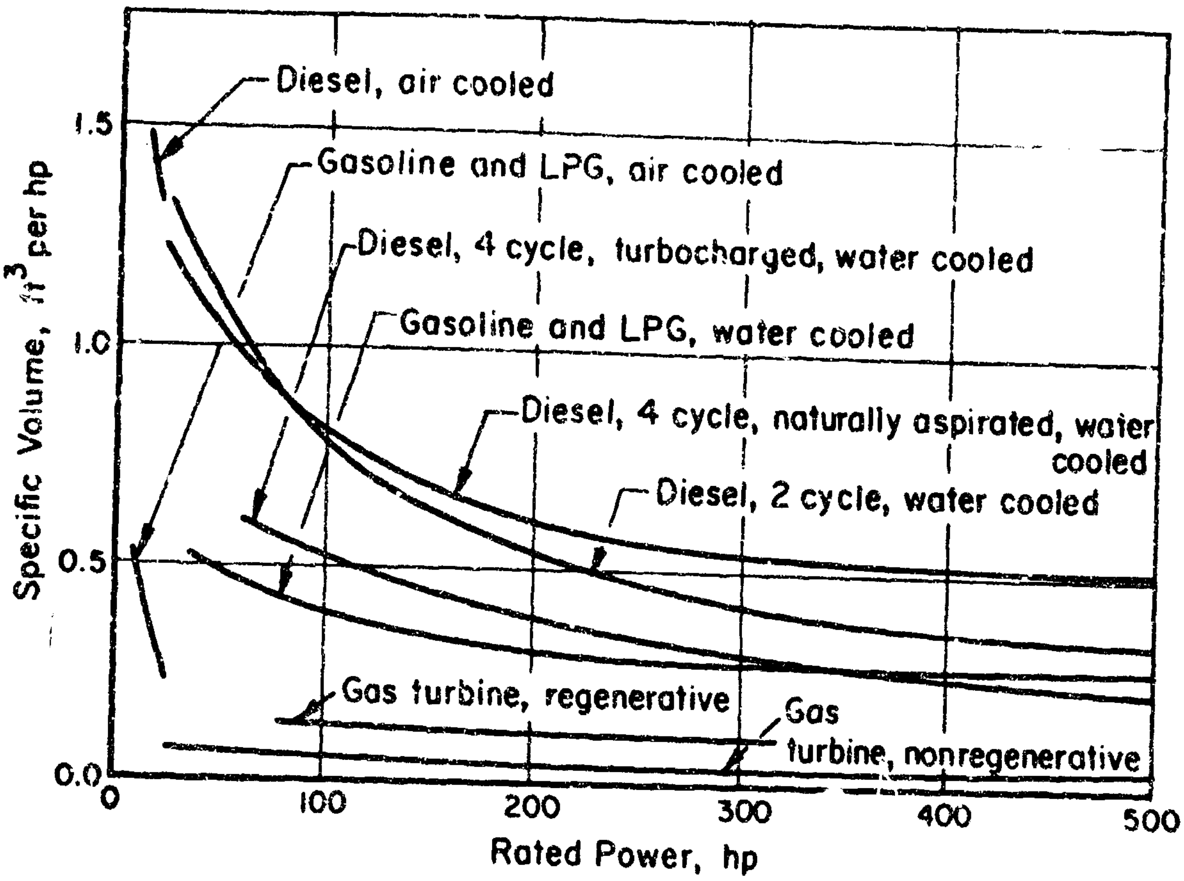


FIGURE 1. SPECIFIC VOLUME OF PRIME MOVERS

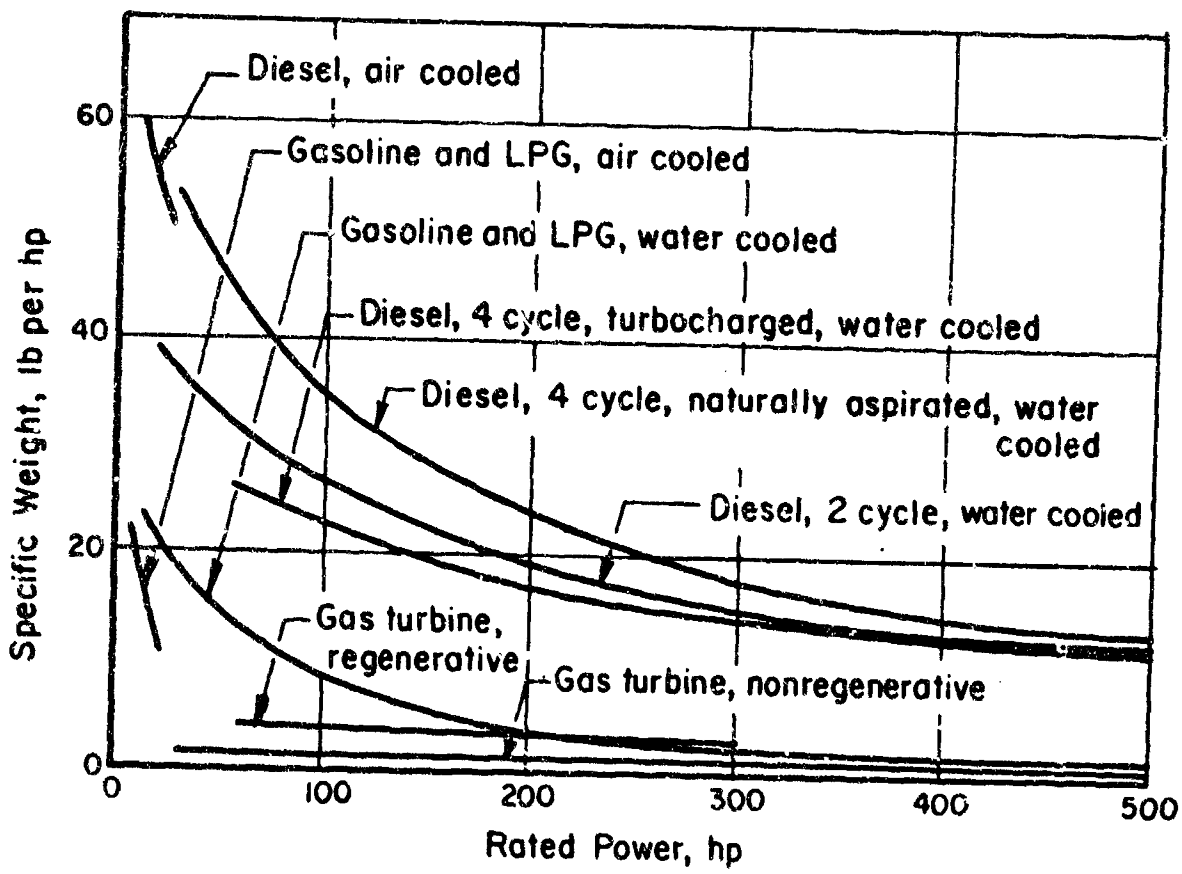


FIGURE 2. SPECIFIC WEIGHT OF PRIME MOVERS

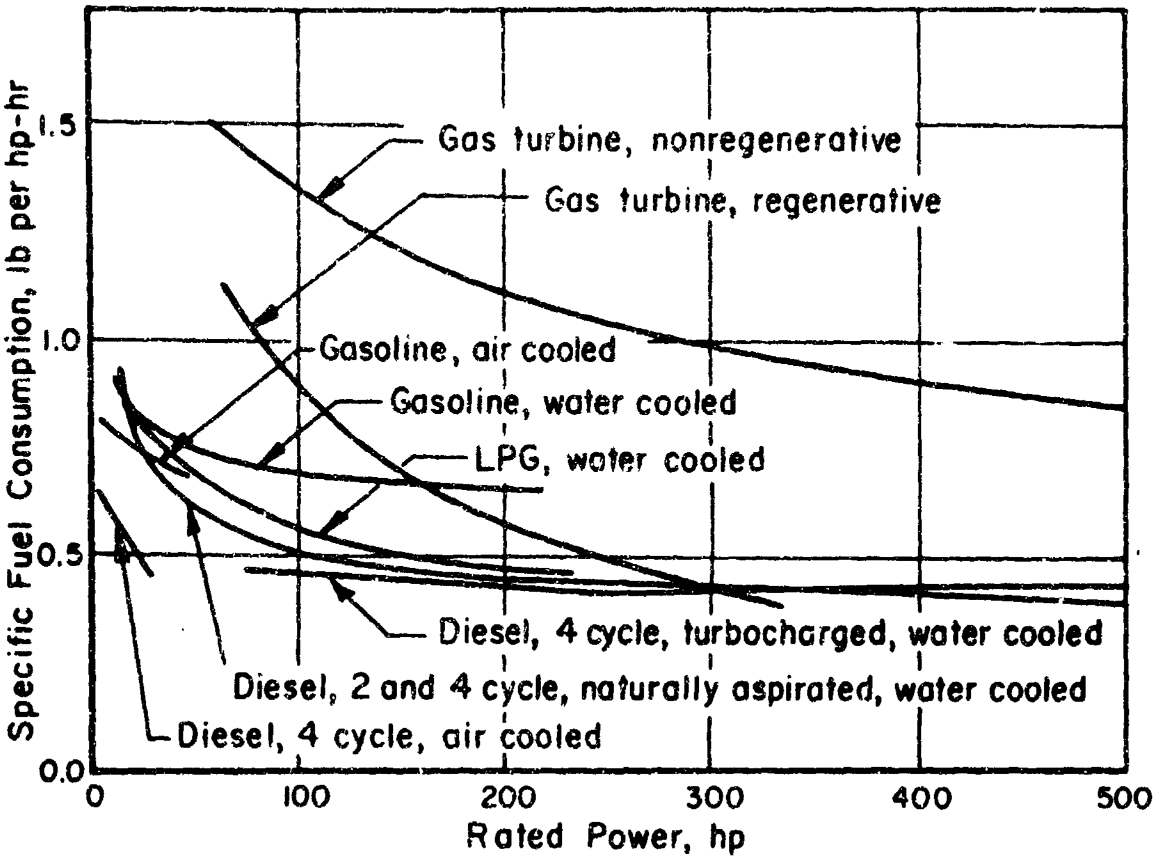


FIGURE 3. SPECIFIC FUEL CONSUMPTION OF PRIME MOVERS

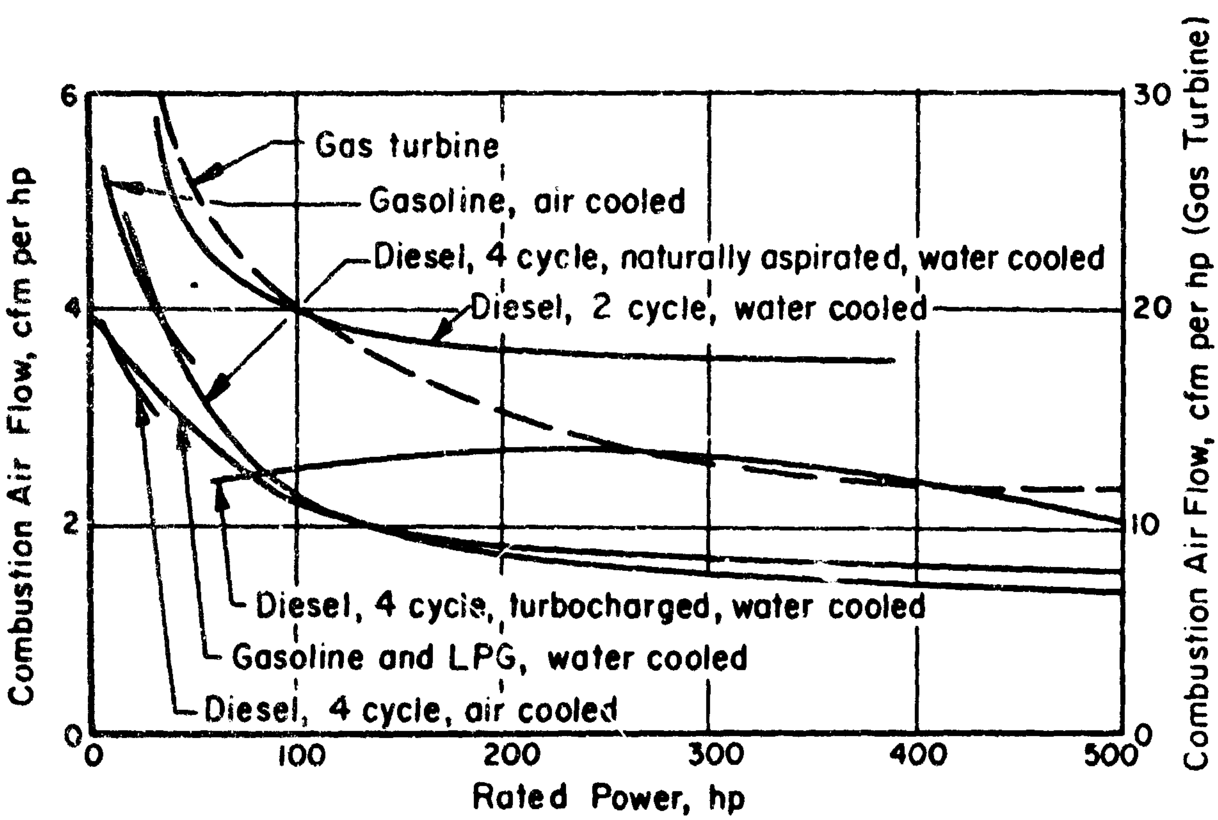


FIGURE 4. COMBUSTION AIR REQUIRED FOR PRIME MOVERS

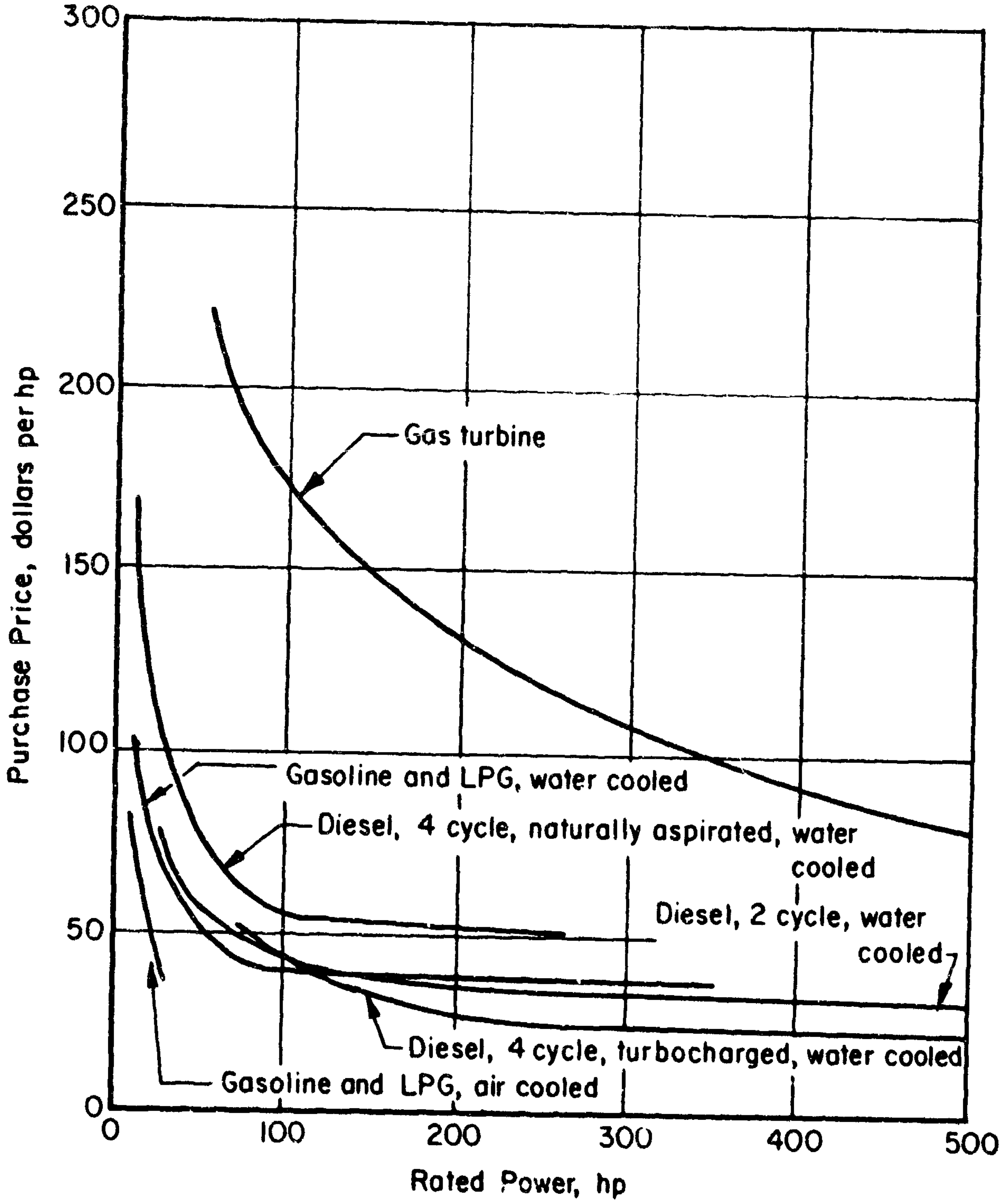


FIGURE 5. APPROXIMATE PURCHASE PRICES OF PRIME MOVERS

Cooling

Direct make-up-water, heat-exchanger, radiator, and ebullient cooling systems were included in this study area. Each was evaluated on the basis of physical characteristics, installation, maintenance, dependability, water or air consumption, and economics.

Table 3 summarizes the results of these evaluations.

Ventilation of the engine room to remove heat lost from the engine and other components by radiation will be essential. These losses may be up to half as much as the jacket water cooling losses. Ventilating the engine room with exhaust air from the occupied section of the shelter may be possible and would be desirable in that a minimum of additional equipment would be required.

Gas Turbines require no jacket water cooling system and are likely to radiate no more heat to the engine room ventilating air than piston engines.

TABLE 3. COMPARISON OF COOLING SYSTEMS

Cooling System	Installation	Maintenance	Water Requirements	Cost
Direct make-up	Simple	Negligible	High	Low
Heat exchanger, without cooling tower or pond	Simple	Negligible	Very high	High
Heat exchanger, with cooling tower or pond	Complex	Moderate	Moderate	Very high
Ebullient (boiling)	Simple	Moderate	Low	Low
Radiator, engine-mounted	Simple	Low	None (air required)	Moderate
Radiator, outside-mounted	Moderately complex	Moderate	None (air required)	Moderately high

Fuel Storage

The storage capabilities of the following fuel types were evaluated in this study area: 90-octane gasoline, straight-run kerosene, No. 2 diesel oil, and liquified petroleum gas (LPG). These are the most commonly used prime mover fuels and can be considered for the purposes of this study as being generally representative of all available commercial fuels.

All fuels deteriorate in storage to some degree, depending on the type of fuel, the storage conditions, and the length of storage time. The primary causes of deterioration in fuels are: evaporation, oxidation, contamination, and polymerization. Any fuel-storage technique which inhibits any of these will increase the storage life of the fuel. Fuel deterioration can be detrimental in reducing ignition and burning qualities of the fuel, increasing corrosion in the fuel system, increasing clogging and fouling in the fuel system, and increasing contamination of the lubricating oil.

Two basically different fuel-storage techniques may be followed: active storage and long-term storage. In an active fuel-storage program a fuel would be replaced or replenished at regular intervals, and the storage system requirements would be relatively uncritical. In a long-term fuel-storage program, the storage system would be designed to preserve the fuel for the longest possible period of time. For an active fuel-storage system a simple vented tank, either above-ground or (preferably) underground, would be suitable. A sealed underground tank would significantly increase the potential storage life of most fuels and would be suitable for a long-term fuel-storage system. Adding a pressurized nitrogen "blanket" to the sealed storage tank would further increase storage life.

Table 4 shows approximate storage life that can be expected with different fuels under different storage conditions.

Waste Heat Recovery

All conventional prime movers convert only part (up to about 1/3) of the total fuel energy supplied to useful shaft work, the remainder being rejected as waste heat to the cooling and exhaust systems or lost as radiation. For piston engines the amounts of waste heat in the coolant and in the exhaust gases are each approximately equivalent to the shaft power output. For regenerative gas turbines the exhaust waste heat is approximately 2-1/2 times the shaft power output.

The waste heat from the cooling system can be fairly easily recovered either: (1) as hot water from a water-to-water heat exchanger or from the jacket water directly, or (2) as low-pressure steam and/or hot water directly from the steam separator tank of an ebullient cooling system. Only about 60 to 80 per cent of the exhaust waste heat is recoverable because of the danger of corrosion in the exhaust system if the gases are cooled below 300 F. Waste heat recovered from the exhaust system can be made available as hot water or as high- or low-pressure steam.

TABLE 4. ESTIMATED STORAGE LIFE OF REPRESENTATIVE FUELS

Storage Condition	Straight-Run Gasoline (About 90 Octane)[c]	Straight-Run Kerosene[c]	Premium-Grade No. 2 Diesel Fuel[c]	LPG[d]
Above-ground vented tank[a]	1 year	3 years	1 year	--
Underground vented tank[b]	2 years	5 years	3 years	--
Underground sealed tank[b]	5 years	8 years	4 years	10 years +
Underground sealed tank with positive nitrogen pressure[b]	8 years	8 years	5 years	--

(a) It is assumed that the fuel temperature varies from 20 to 100 F throughout the year, and that the amount of fuel stored is at least 500 gallons.
(b) It is assumed that the fuel temperature varies from 40 to 80 F throughout the year, and that the amount of fuel stored is at least 500 gallons.
(c) It is assumed that for cold starting, the operator manually sprays ether aerosol into the air intake. That is, the fuel is not considered useless on the basis of poor cold starting due to the loss of the lower-boiling-point components.
(d) In all instances, LPG is stored in a sealed tank.

Recovery of waste heat for use in providing a hot-water supply or for space heating is practical when there is a reasonable demand for these services in the shelter. It is more advantageous to recover waste heat from the cooling system because: (1) less extra equipment is needed, (2) the exhaust waste heat is easily rejected from the shelter, and (3) there is a danger of overheating and corrosion with an exhaust heat-recovery system that does not exist with a cooling water heat-recovery system.

Recovery of waste heat for shaft power is not practical because of the cost and complexity of the extra equipment required.

Power-Transmission Systems

Mechanical, electric, hydraulic, and pneumatic power-transmission systems were included in this study area. Each was evaluated on the basis of physical characteristics, over-all efficiency, installation and maintenance requirements, reliability, safety and human comfort, and economics.

Table 5 summarizes the results of these evaluations. The mechanical, hydraulic, and pneumatic systems all require a small electric generator to supply the lighting and communications requirements of the shelter. The electric power-transmission system has an advantage in that it can be more easily integrated into a commercial power system for independent exercising of individual components, for potential use of the shelter space during the stand-by period, and for use of commercial power if it is available in an emergency.

TABLE 5. COMPARISON OF POWER-TRANSMISSION SYSTEMS

Parameter	Power Transmission System			
	Mechanical	Electric	Hydraulic	Pneumatic
Installation	Simple to complex	Simple	Complex	Complex
Maintenance	Negligible	Low	Low	Low
Dependability	Excellent	Good	Good	Good
Availability	Excellent	Excellent	Good	Good
Efficiency, %	90 +	76	66	14
Cost	Very low	Low	High	High

System Mountings and Drives

Five items to be considered in designing or selecting mountings and drives are: (1) alignment, (2) vibration, (3) piping connections, (4) load characteristics, and (5) blast effects. Mounting and drive components should be selected to prevent annoying or destructive vibration from developing within the mounted components and from being transmitted to or from other parts of the shelter.

Applicable mounting techniques are foundation mounting or skid mounting, and flexible or rigid mounting. Flexible skid mounting is preferable for community shelter auxiliary power systems as it would result in acceptable component alignment, minimum vibration transmission, and low installation cost.

Of the applicable drive techniques, a flexible direct-coupling drive is preferable for community-shelter auxiliary-power systems as it would be compatible with skid mounting and would function with a minimum of vibration transmission. A clutch is necessary when the driven-component starting torque is high or when the prime mover and driven component shafts must be rotated independently.

Noise and Vibration

Reliable and useful data on the noise generation characteristics of prime movers are not presently available. However, from general experience and limited test measurements made at actual installations and in the Battelle test facility, it is apparent that some noise control measures will be required with most prime movers. However, the noise problem should not be difficult to solve.

The chief sources of noise from prime movers are vibrating surfaces and aerodynamic pulsations. Control of this noise can be effected in the following ways:

(1) Careful selection of the prime mover and its auxiliary equipment.
(2) Provision of barriers between the prime mover and the shelter occupied space.
(3) Reduction or elimination of transmission of vibration from the source.
(4) Use of sound-absorbing materials on the walls and ceiling of the occupied shelter space.
(5) Providing "sound trap" air ducts when noise source area and occupied space must be connected for ventilation.

The experimental work in the Battelle test facility indicates that thick walls, complete closure, and sound-trap ducting would be effective in reducing the noise reaching the occupied space to an acceptable level. It was also noted that no special vibration isolation equipment would be necessary to prevent annoying engine vibration from reaching the occupied space. The normal construction of underground shelters would be adequate to damp the vibration.

Stand-By Maintenance

Two general approaches to stand-by maintenance were considered in this study: dynamic maintenance and static maintenance. Dynamic maintenance involves frequent and periodic exercising and inspection and is employed almost universally for conventional emergency stand-by power systems. Static maintenance involves long-term storage with infrequent exercising and inspection.

Dynamic maintenance is essential if instant starting and load assumption is required in the event of commercial power failure. The procedures for dynamic maintenance are fairly well established from experience. A minimum frequency of once every six weeks, and a 1 to 2 hour full-load run are advisable. Periodic changes of lubricants and coolant are necessary to prevent damage or inoperability due to time-dependent deterioration of these liquids.

Static maintenance may necessitate allowing a period of 1 or 2 hours for reactivation of the equipment. During reactivation a community shelter could be occupied with emergency lighting and communications supplied by long-shelf-life batteries. A satisfactory static maintenance technique could reduce the long-term maintenance costs of a community shelter program a significant amount. The static maintenance program would require maintaining a low (30 to 35 per cent) humidity level in the shelter, keeping liquids that tend to deteriorate away from components which may be easily fouled or otherwise rendered inoperable, and coating critical surfaces with a long-life preservative.

Static maintenance techniques presently in use, such as "mothballing" of surplus military equipment, are not directly applicable to the community shelter because of cost and the time required (a minimum of several days) to reactivate. However, the available experience and proven techniques are sufficiently encouraging to indicate the potential feasibility of the approach.

Demonstration Unit

A 20-kw diesel engine-generator set was set up in an underground test facility to experimentally verify the parameters developed in this study in a simulated community shelter environment. The same unit was also used to evaluate the performance of simple exhaust heat-recovery systems.

Figure 6 is a sketch of the test facility with the engine-generator set installed. An air-cooled radiator is mounted outside the shelter and the ventilating air for the engine room is supplied through a "sound trap" duct. The entrance hallway was used to simulate the shelter occupied space.

As part of the experimental studies, two test runs were made to determine the system heat balance and to measure the effect of insulating the exhaust system. For one run the exhaust system was not insulated and for the other run 1-inch thick insulation was applied to the exhaust manifold and tubing in the engine room. The test results effectively demonstrated that insulating the exhaust system reduced the amount of heat rejected to the cooling system, ventilating air, and walls, thus reducing the shelter heat sink requirements.

The results of the demonstration program verified the validity of the parameters developed in the study.

Because commercial exhaust heat-recovery equipment is costly, several simple configurations were studied to determine their effectiveness. About 80 per cent of the heat in the exhaust gases is recoverable without encountering exhaust system corrosion. A straight 3/4-inch-diameter brass tube installed in the exhaust system had a recovery efficiency of 17 per cent. A coiled tube and a straight-finned tube both 3 feet long each had an efficiency of 36 per cent. From these studies it can be concluded that simple, inexpensive exhaust heat-recovery systems are effective and practical.

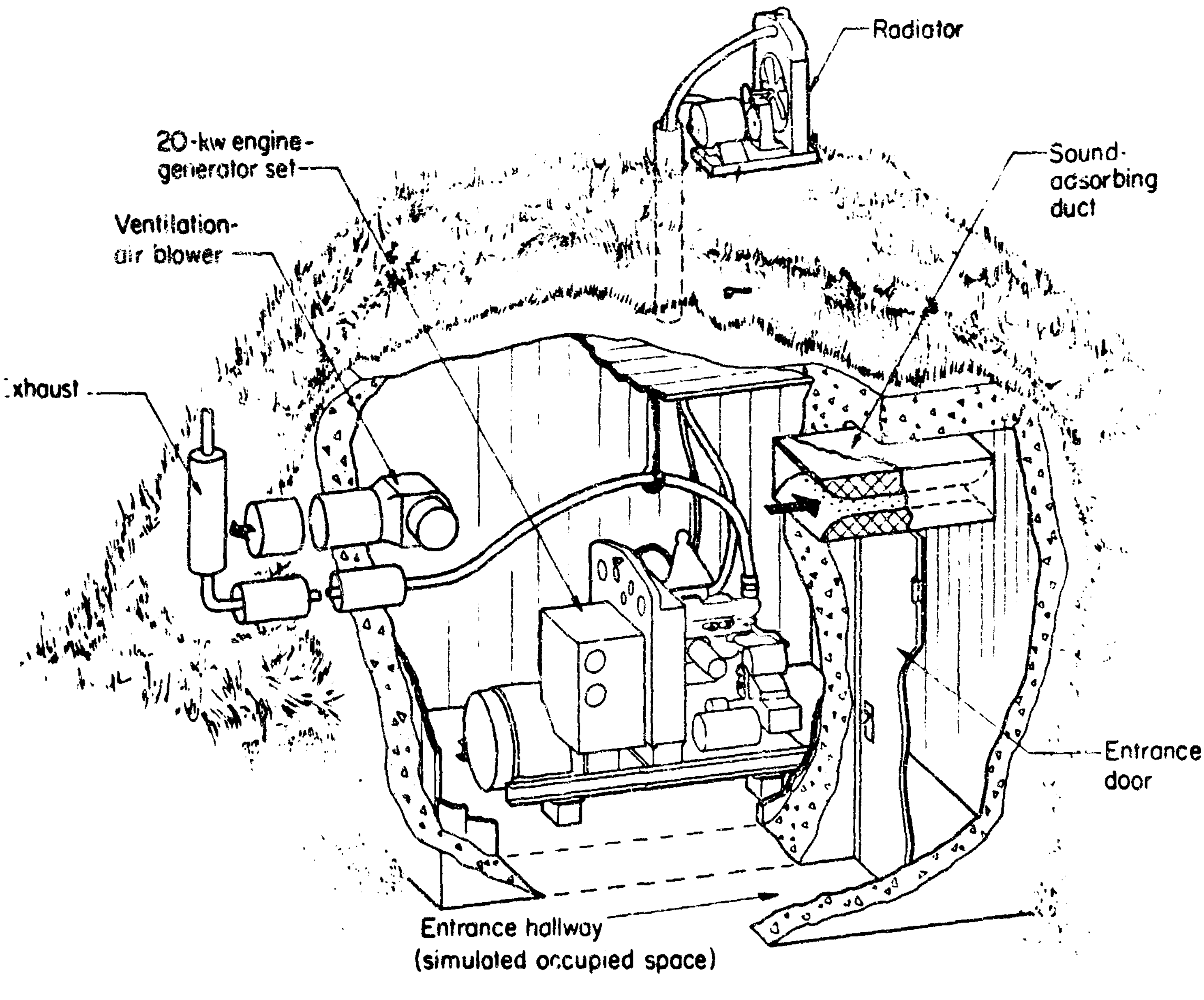


FIGURE 6. UNDERGROUND TEST FACILITY

FUTURE RESEARCH NEEDS

This research study disclosed a number of technological areas in which the presently available equipment and experience do not adequately meet the needs of community shelters. These areas are:

(1) Closed-cycle internal-combustion engine operation.
(2) Manual energy-storage starting systems.
(3) Long-term preservation and storage of fuels.
(4) Long-term preservation and storage of equipment.
(5) Low-cost waste-heat recovery equipment.

Because of the importance of these subjects in the over-all community shelter program, Battelle recommends that further research be conducted to determine feasible and practical solutions to the unsolved problems.

Closed-cycle operation of an internal-combustion engine prime mover would involve recirculating the exhaust gases through the engine after removal of noncombustible solids. Sufficient oxygen for combustion would be added to these recirculated exhaust gases before they were allowed to enter the engine. A portion of the exhaust gases, equal to the oxygen added, would be discharged from the system to the atmosphere. Closed-cycle operation would permit the prime mover to develop partial or full power during periods when it would be necessary to completely isolate the shelter from the external environment. Investigations of closed-cycle diesel engine operation for submarine use during World War II and for mine use more recently would provide background for further investigations.

Manual energy-storage starting systems are discussed in this report. The feasibility study which was conducted indicated that a flywheel or similar type of manually operated energy-storage starting system could be developed which would be competitive with conventional starting systems on the basis of first cost. The manual system would be simple, maintenance free, and extremely reliable.

Long-term preservation and storage of fuels has evidently received little attention. The information which is available is incomplete and inconclusive for the community shelter system application However, there is evidence that most fuels can be stored for long periods of time with only minor modifications to presently used storage equipment and techniques and to the chemical composition of the fuel.

Long-term preservation and storage of equipment as presently practiced is costly and requires an excessive amount of time for reactivation. However, as in the case of storage of fuel, the present state of knowledge and experience strongly indicates that techniques could be developed that would combine low cost with the required state of readiness for community shelters.

Low-cost waste-heat recovery equipment is not commercially available at present. A brief experimental study conducted during this project demonstrated the feasibility of adapting simple heat exchanger equipment to recover waste heat from the engine exhaust gases. Further study, including both design and experimental phases, would provide the information necessary to construct satisfactory and low-cost equipment for the recovery of exhaust and cooling-system waste heat.

PRIME MOVERS

The study of prime movers was limited to conventional equipment, four basic types being investigated: compression-ignition (diesel) engines, spark-ignition engines, gas turbines, and steam power plants. These four basic prime mover types were considered on the basis of performance and control characteristics, installation and maintenance requirements, safety and human comfort aspects, and economics.

All of the data presented in this section of the report pertain to prime movers in the 5- to 500-hp size range. Most compression-ignition and spark-ignition engines in this size range are available as production engines and as self-contained power sources. Larger piston engines are available, but mostly on a custom-order basis in relatively limited quantities; consequently, the cost per horsepower output will tend to be higher than for the smaller engines. Installation of the larger piston engines will also be more costly and difficult because of their physical size. The performance characteristics of the larger engines will be very similar to those of the smaller engines of the same type.

Large gas turbines may be more attractive, comparatively, than large piston engines because gas turbines are inherently compact and simple machines. Consequently, for a given power output they are much smaller and lighter than the same size piston engines and, therefore, they are much easier to install and service than a piston engine. Large gas turbines are so, at the present time, significantly more efficient than small ones because of the proportionately greater development effort which has gone into their design.

Compression-Ignition Engines

Compression-ignition engines, more commonly referred to as diesel engines, a[illegible] [illegible]nerally heavy-duty power sources used for continuous duty where low-speed lugging ability is desirable. Diesel engines come in a great many different forms: air and water cooled, two and four cycle, naturally aspirated, supercharged, and turbocharged.

Figures 7 and 8 show typical specific volume and weight data for diesel engines up to 500 hp output. The particular diesel-engine types represented by the curves of Figures 7, 8, 9, and 10 were selected because they are representative of all of the available types and are the most suitable for stand-by auxiliary power system use. For all types shown, the specific volume and weight are significantly greater in the smaller power sizes. The turbocharged and two-cycle diesel engines are considerably smaller and lighter than the naturally aspirated diesel engines.

Diesel engines up to 500 hp are designed to operate at rated speeds from 1200 to 2400 rpm. Industrial diesel engines in this power range are available as follows:

Two-cycle, water-cooled, supercharged	33-500 hp
Two-cycle, water cooled, turbocharged	140-500 hp
Four-cycle, air-cooled, naturally aspirated	6-23 hp
Four-cycle, water-cooled, naturally aspirated	4-500 hp
Four-cycle, water cooled, turbocharged	70-500 hp

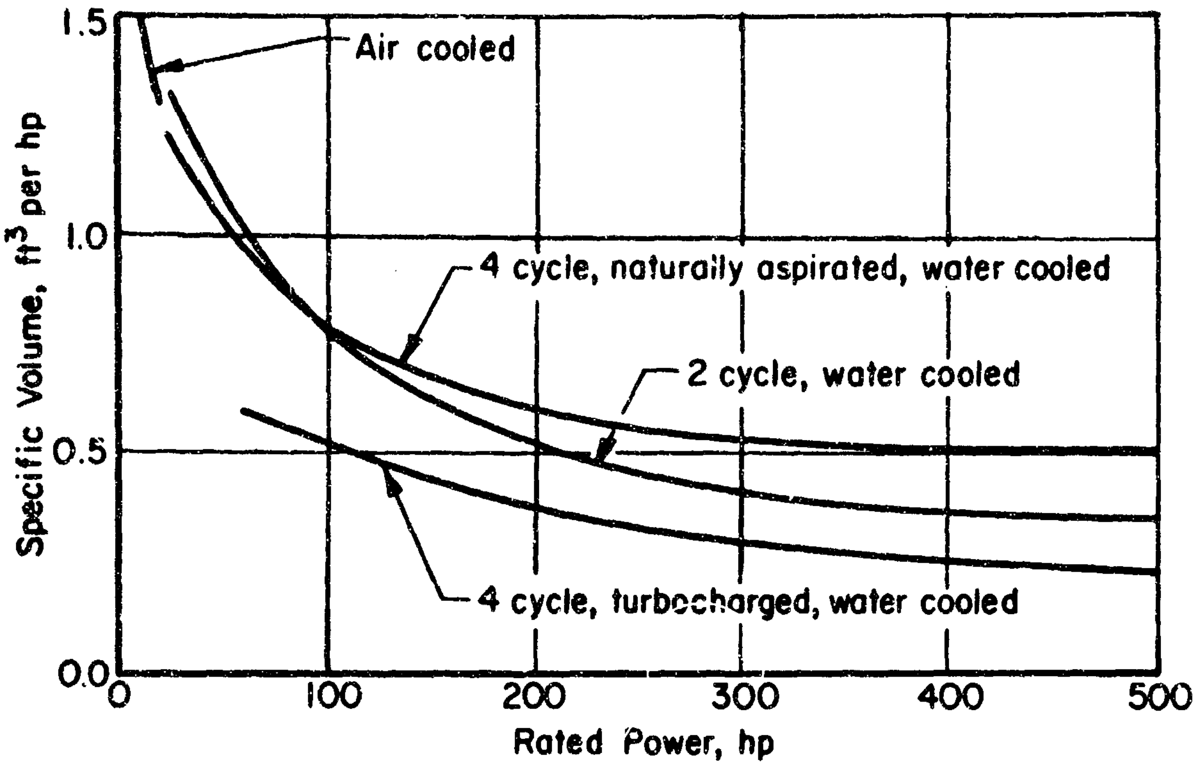


FIGURE 7. SPECIFIC VOLUME OF DIESEL ENGINES

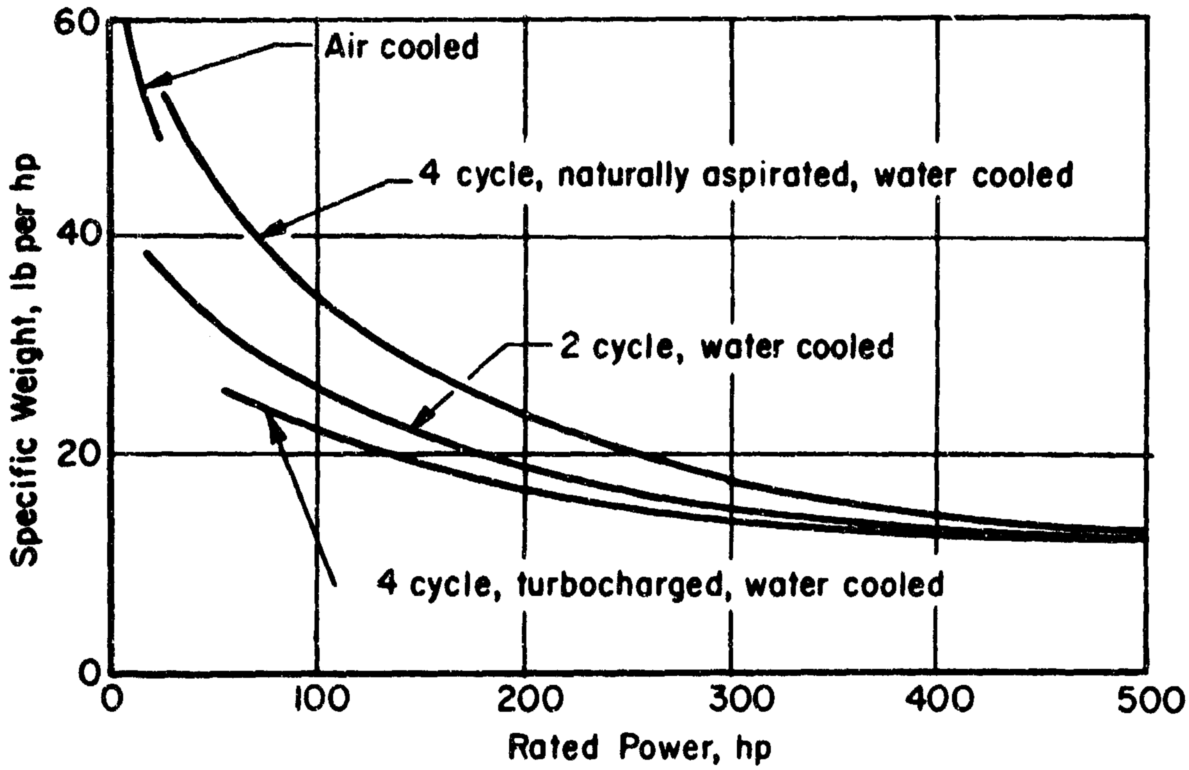


FIGURE 8. SPECIFIC WEIGHT OF DIESEL ENGINES

Performance and Control

Figures 9 and 10 show specific fuel consumption and combustion air requirements for diesel engines in the 5- to 500-hp range. Highest fuel consumption and combustion air requirements are associated with the lower power outputs. The shape of the curve for the combustion air requirement of the turbocharged diesel engine is different because of sensitive interactions between the turbochargers and the engines. For economy only a few different turbocharger models are used with a wide range of engine sizes; consequently, there is bound to be some slight mismatching. In addition, as more power is extracted from an engine of a given size by the use of turbocharging, more excess combustion air is required to maintain acceptable peak combustion-chamber temperatures, particularly to prevent piston failure.

Diesel engines are sensitive to intake and exhaust conditions. For instance, the maximum power rating of a diesel engine should be decreased about 1 per cent for each 10-degree temperature rise in the intake air above 60 F, and 4 per cent for each 1,000 feet of altitude (i.e., a reduction in pressure of 14 inches of water from normal pressure in the intake system). Two-cycle diesel engines are very sensitive to exhaust back pressure, four-cycle naturally aspirated diesel engines are less sensitive; and turbocharged diesel engines are relatively insensitive.

Diesel engine manufacturers frequently publish three power output ratings for their engines. These are: continuous-duty rating, intermittent-duty rating, and maximum rating. The continuous-duty rating represents a power level at which the engine should be able to operate continuously and still achieve its normal service life. An intermittent-duty rating is given in recognition of the fact that many applications will not require full engine power at all times. The intermittent-duty rating is generally about 10 per cent higher than the continuous-duty rating, and when used for applications where full engine power is not required at all times will not result in any reduction in service life. The maximum rating is about 20 per cent above the continuous-duty rating and should be used only for occasional momentary high loads.

These power output ratings are conservative to some degree, depending upon the manufacturer of the engine and are based on a normal service life of 8,000 to 10,000 hr between major overhauls. It is probable that most well-designed and well-constructed diesel engines could be operated continuously at the intermittent-duty rating with only one detrimental effect, reduced service life. If adequate cooling were provided, it is even possible that a service life in excess of 500 hr could be relatively assured even with continuous operation at or near the maximum power rating.

Speed and load control systems for diesel engines are fairly simple and reliable as long as the regulation requirements are not too strict. A plus or minus 5 per cent speed variation, which would probably be quite acceptable for a community shelter auxiliary power system, would be relatively easy to achieve. Manual controls usually supplied with diesel engines include an on-off switch, a starting button, and governor and throttle control cables.

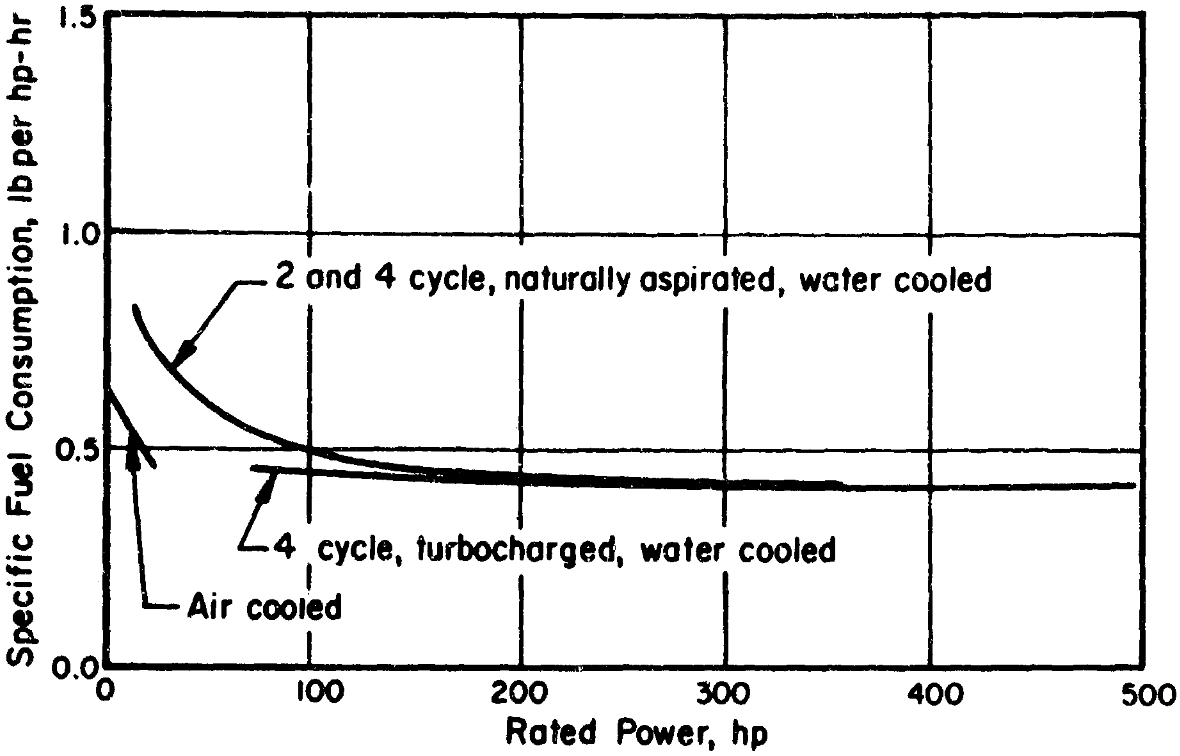


FIGURE 9. SPECIFIC FUEL CONSUMPTION OF DIESEL ENGINES

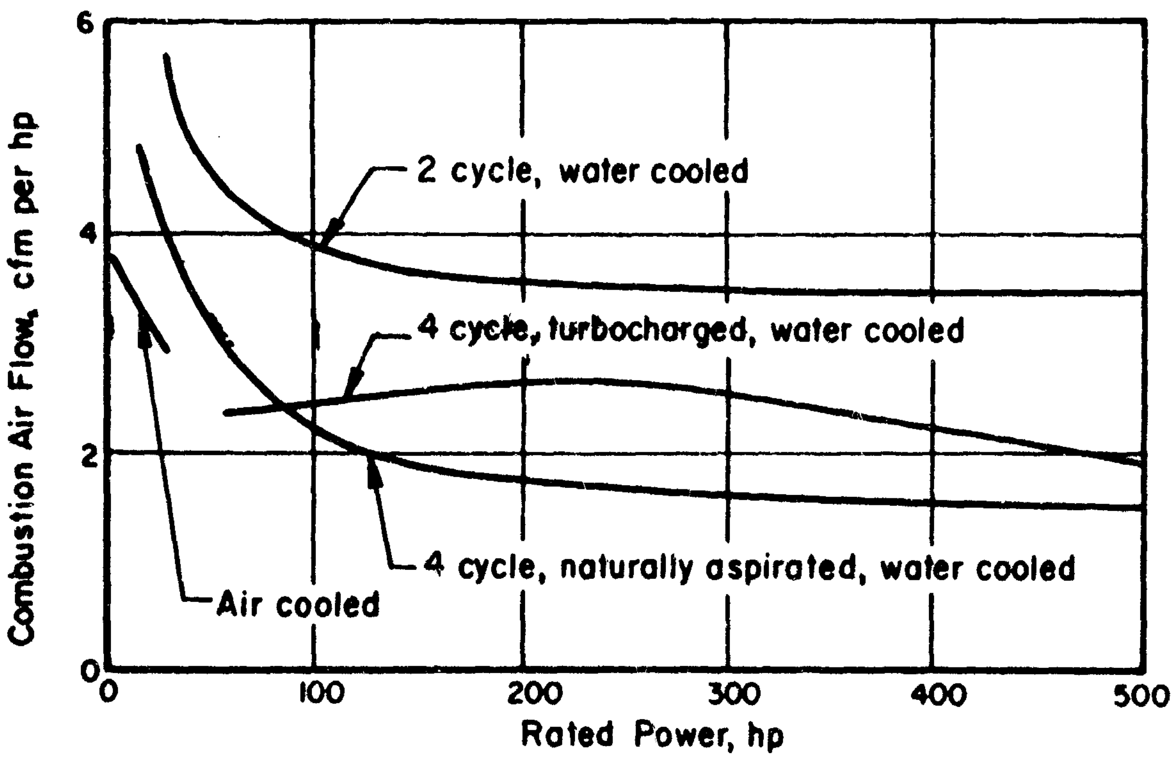


FIGURE 10. COMBUSTION AIR REQUIRED FOR DIESEL ENGINES

Installation

There are four major factors which must be given consideration when a diesel engine is installed for emergency stand-by power. These are mounting, ventilation, fuel supply, and exhaust. All diesel engines should be mounted on vibration dampers and preferably on a base or on blocks at a level higher than the floor level for convenience in servicing and maintenance. For the larger engines, the base should be isolated from the rest of the engine room floor. At least 2 feet of space should be provided all around the engine. The engine should be located convenient to the fuel supply, exhaust outlet, and ventilating air outlet.

Sufficient ventilation air should be provided to the engine room for combustion and for carrying away radiated heat. If the engine is air cooled, or radiator cooled with the radiator located in the engine room, additional ventilation air must be provided. The engine should be provided with its own combustion air filter, hence, filtering of the ventilation air would not be necessary. Fall-out particles which may pass into and through the engine and cooling system are not expected to affect the performance. Both the inlet and the discharge ventilation air openings above the ground level should be located so that the possibility of clogging from fallen debris is minimized. Inside the engine room, the ventilation air inlet and discharge outlet should be on opposite walls. If an engine-mounted radiator is used, it is advisable to provide a duct between the radiator and the discharge opening in the engine room wall to prevent recirculation of the cooling air. This duct should contain a flexible section so that engine vibration is not transmitted to the wall and so that the danger of fatigue failure of the duct will be minimized.

It is not generally acceptable to allow the fuel to be supplied to the engine from the main tank by gravity feed, because of the danger that a leak anywhere in the system might permit the entire fuel supply to drain into the engine room. A "day" tank mounted directly on the engine is frequently used to assure that the fuel will be immediately available to the engine on starting. If possible, the main fuel tank should be located close to the engine room and slightly below the level of the fuel pump on the engine. A flexible section should be provided in the fuel line to the engine to reduce the transmission of engine vibration and to avoid fatigue failure in the fuel line. If the main fuel tank must be located above the engine, a float tank should be used between the main tank and the engine. This should be located below the level of the engine fuel pump. With such an arrangement, fuel runs by gravity from the main tank to the float tank where a constant level is maintained by a float-valve system. The engine fuel pump draws fuel from the float tank. If the fuel tank must be located at a greater distance from the engine room than the engine fuel pump is capable of pumping the fuel, a transfer tank and transfer fuel pump must be provided between the main fuel tank and the engine. The transfer pump is needed to pump fuel from the main tank to the transfer tank. The engine fuel pump draws fuel from the closer transfer tank.

The exhaust pipe from the engine to the outside should be adequately sized, and should be well constructed, without any leaks, and well supported. The exhaust pipe should slope upward from a point near the engine and a condensate trap should be provided at that lowest point. Condensate must not be allowed to drain back into the engine. A flexible section should be provided in the exhaust line to reduce transmission of vibration and to avoid fatigue failure of the exhaust pipe. The exhaust pipe should be no closer than 9 inches to any combustible material. The exhaust outlet outside the shelter should be so located with respect

to the ventilation air inlet that the possibility of exhaust products being drawn back into the shelter is negligible. If the shelter is located in a heavily populated area and frequent exercising of the equipment is planned, an inexpensive muffler probably should be provided. It is generally recommended that mufflers be located as close to the engine as possible to reduce corrosion damage to the muffler from condensed products of combustion.

Maintenance

Consideration of the anticipated short-time operation of community shelter power sources when compared with the relatively long service life of diesel engines leads to the conclusion that there should be no maintenance requirements due to running of the engine. Consequently, the only maintenance problems that should arise are those due to long storage time. There probably will be maintenance problems associated with the fuel system, the lubrication oil system, the cooling system, and the starting system. The fuel system, cooling system, and starting system requirements will be discussed in later sections of this report. Operational lubricating oils will tend to form sludge and gum when stored and little used for long periods of time. If a regular schedule of frequent exercising is planned, it would be desirable to test a sample of the lubricating oil at least once every two or three months for signs of deterioration. If the power source is not to be exercised frequently, it would be feasible to replace the operational lubricating oil with a preservative oil which would have a much longer storage life. A few spare parts such as fuel injectors, fuel pumps, fuel and oil filter elements, and water pumps would be handy to have. A semiskilled technician with proper instruction could replace any of these parts in the event of a failure during an emergency.

Safety and Human Comfort

The noise level in the engine room, particularly with the larger installations, will be fairly high. However, it should not be difficult to reduce the fraction of the noise which reaches the occupied spaces of the shelter to an acceptably low value. Simple and inexpensive approaches to accomplishing this include acoustic treatment of wall and ceiling surfaces inside the engine room, double-wall construction of the engine room enclosure, seals around all doors and other access openings between the engine room and the occupied spaces, and separation of the occupied space from the engine room by buffer spaces such as equipment and supply rooms.

Diesel fuel is not highly flammable and, consequently, it does not present a very serious fire hazard. Diesel exhaust gases are objectionable from the standpoint of odor, but they are not toxic. Most diesel engines are provided with high water temperature and low water pressure cutoff devices which sound an alarm and/or cut off the engine when a previously established maximum operating point has been reached. An overspeed cutoff device can also be provided.

Economics

Figure 11 shows approximate purchase prices of diesel engines in the 5- to 500-hp range. These data were obtained for engines supplied for electric generator service. This is a highly competitive field. The same trend of higher specific values for smaller size units appears in these curves as in the curves of the previous two figures. The cost of installation of these engines can be expected to add between 10 and 30 per cent to the purchase price figures shown in Figure 11. The larger the engine, the greater will be the cost of installation. Fuel cost for a diesel engine will be quite nominal for the expected two-week emergency period. For example, a two-week supply of No. 2 diesel fuel for a 100-hp engine would cost about $300.

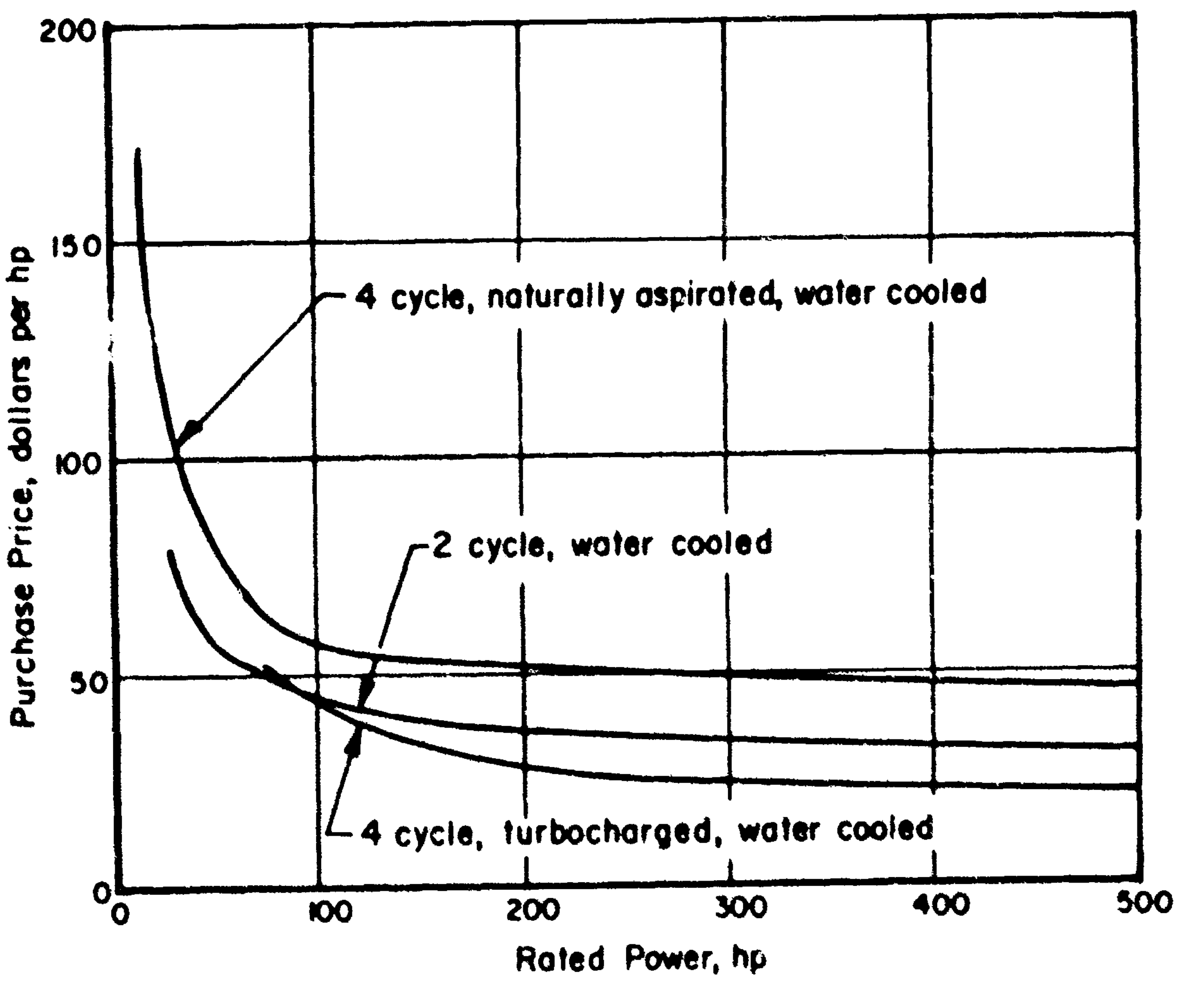


FIGURE 11. APPROXIMATE PURCHASE PRICES OF DIESEL ENGINES

Spark-Ignition Engines

Spark-ignition engines are carbureted engines commonly using gasoline as a fuel. However, they are also available for gas operation on natural gas, manufactured gas, or liquified petroleum gas (LPG). The familiar automobile engine is a gasoline spark-ignition engine. Industrial gasoline (or gas) engines, however, are more conservatively designed and rated than automobile engines for longer service. Spark-ignition engines are available in as many different forms as are diesel engines, i.e., air and water cooled, two and four cycle, naturally aspirated, supercharged, and turbocharged. Two-cycle spark-ignition engines are not often used in industrial service in sizes greater than about 10 or 15 hp because of their relatively short life and poor speed regulation. Supercharged and turbocharged spark-ignition engines are also not very widely used in industrial service because of associated high-temperature problems.

Figures 12 and 13 show typical specific volume and weight data for gasoline and LPG engines in the 5- to 500-hp range. These data represent only the four-cycle naturally aspirated air- and water-cooled spark-ignition engines, as the other types are not considered applicable for stand-by auxiliary power systems because of the problems of short life, poor speed regulation, and high temperatures mentioned in the preceding paragraph.

Spark-ignition engines up to 500 hp are designed to operate at rated speeds from 1200 to 3600 rpm. Industrial gasoline engines in this power range are available as follows:

Two-cycle, air-cooled, naturally aspirated	1-10 hp
Four-cycle, air-cooled, naturally aspirated	1-70 hp
Four-cycle, water-cooled, naturally aspirated	3-500 hp

Industrial LPG engines in this power range are available as follows:

Two-cycle, water-cooled, supercharged	350-500 hp
Four-cycle, air cooled, naturally aspirated	2-70 hp
Four-cycle, water-cooled, naturally aspirated	4-500 hp
Four-cycle, water-cooled, turbocharged	225-500 hp

Performance and Control

Figures 14 and 15 show specific fuel consumption and combustion-air requirements for spark-ignition engines in the 5- to 500-hp range. These data show the LPG engine to be slightly more economical in fuel consumption than the gasoline engine, particularly in the larger sizes.

Spark-ignition engines are about as sensitive to intake and exhaust conditions as diesel engines, and they require approximately 1 per cent derating for each 10-degree temperature rise in the intake air above 60 F and 4 per cent derating for each 1,000 feet of altitude.

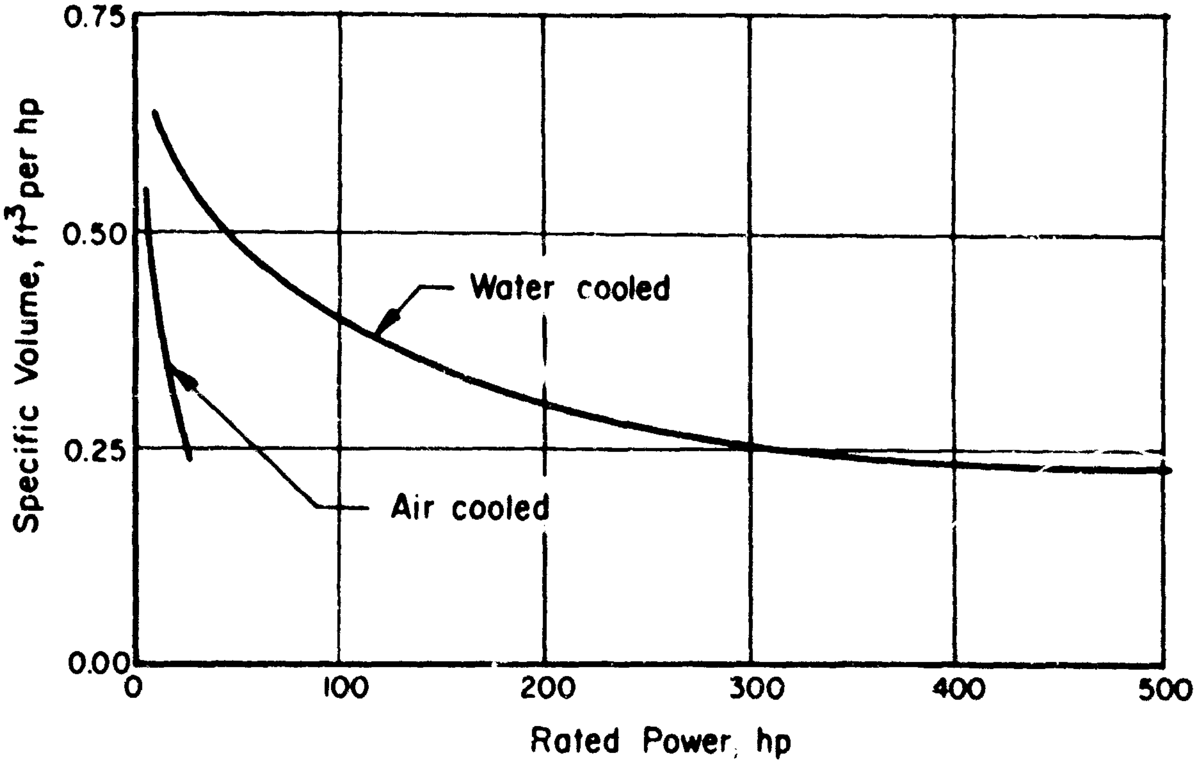


FIGURE 12. SPECIFIC VOLUME OF GASOLINE AND LPG ENGINES

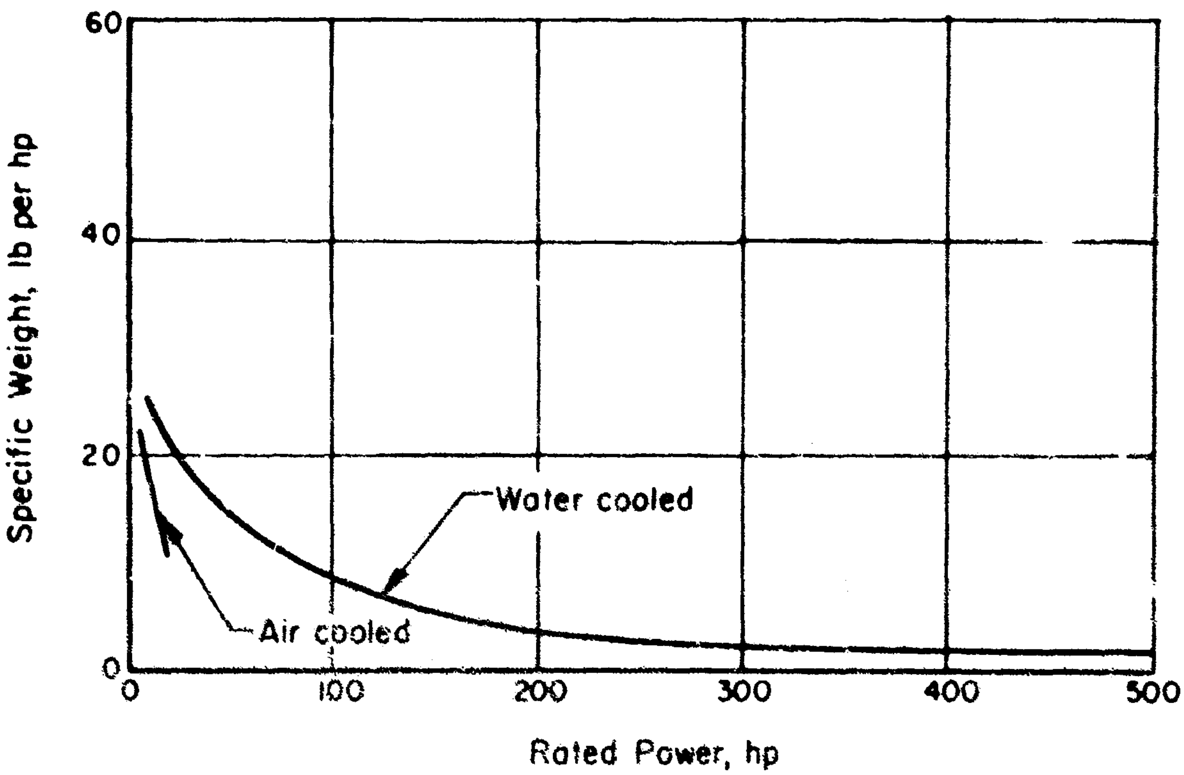


FIGURE 13. SPECIFIC WEIGHT OF GASOLINE AND LPG ENGINES

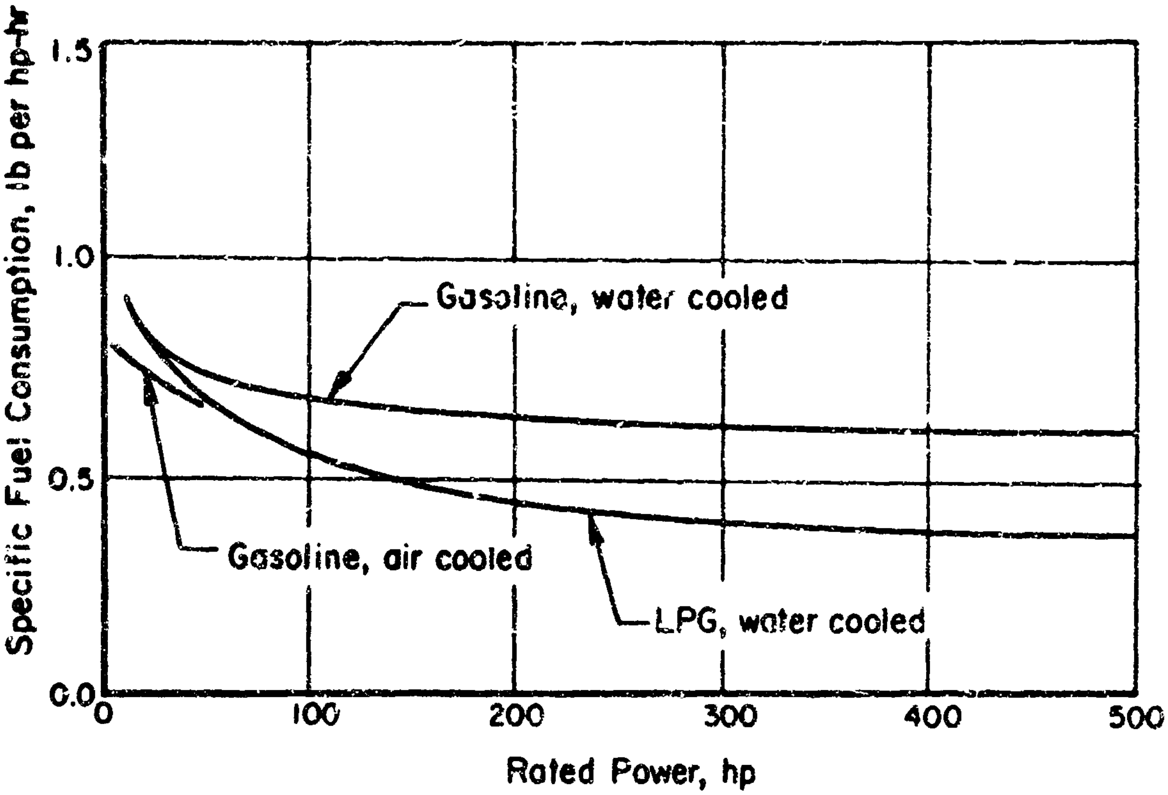


FIGURE 14. SPECIFIC FUEL CONSUMPTION OF GASOLINE AND LPG ENGINES

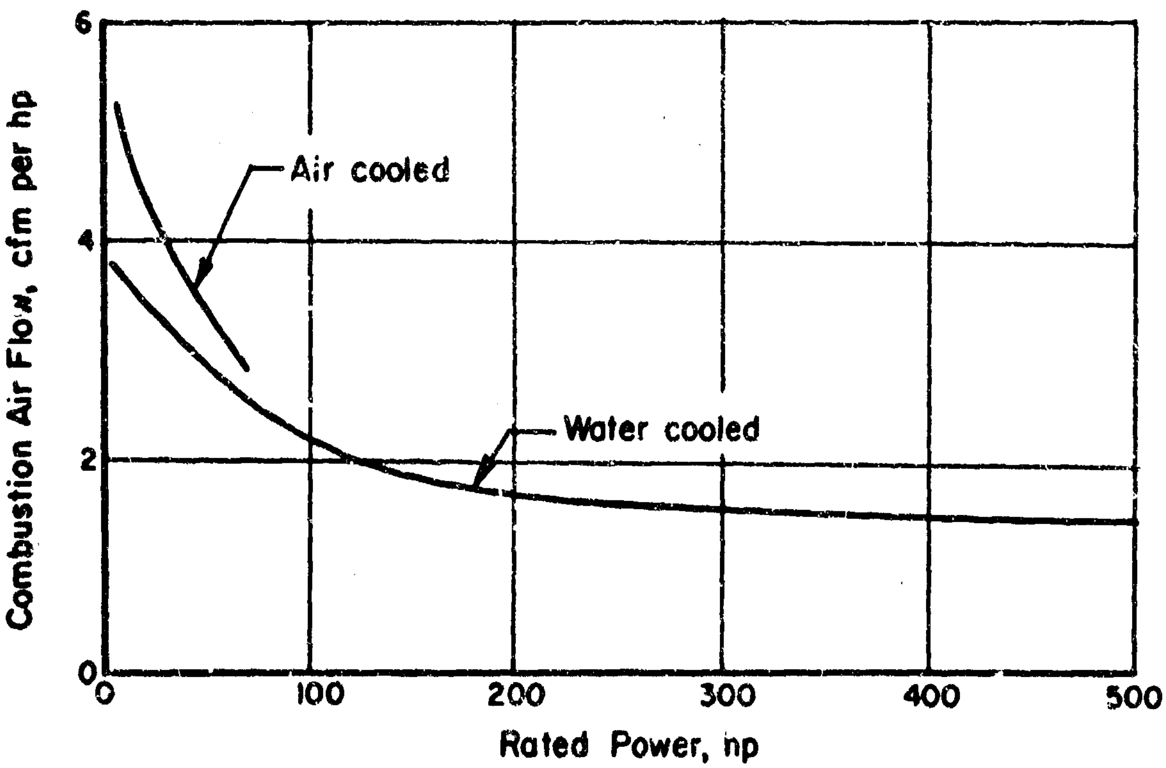


FIGURE 15. COMBUSTION AIR REQUIRED FOR GASOLINE AND LPG ENGINES

Spark-ignition engine manufacturers generally publish three power output ratings for their engines. As in the case of the diesel engines, these are: continuous-duty rating, intermittent-duty rating, and maximum rating. These power output ratings for spark-ignition engines may not be quite as conservative as those for diesel engines; however, it is probable that a well-designed and adequately cooled spark-ignition engine could be operated continuously at the intermittent-duty rating with only a reduction in the total service life.

Speed and load control systems for spark-ignition engines are simple and reliable as long as the regulation requirements are not strict. A speed variation of ± 5 per cent would be relatively easy to achieve. Manual controls provided with spark-ignition engines include an on-off switch, a starting button, a throttle control, a choke control, and sometimes a spark-advance control.

Installation

Installation of spark-ignition engines for emergency stand-by power should follow the same pattern as for diesel engines. All but the very smallest engines should be mounted on vibration dampers and on a base or blocks at a level higher than the floor level for convenience of servicing and maintenance. For the larger engines, the base should be isolated from the rest of the engine room floor. At least 2 feet of space should be provided all around the engine. The engine should be located in the engine room convenient to the fuel supply, exhaust outlet, and ventilating air outlet.

The ventilation air requirements for spark-ignition engines are, if anything, greater than those for the diesel engines. Radiation and miscellaneous heat losses from spark-ignition engines tend to be higher relative to the power output than with diesel engines. The engine room ventilation air does not have to be filtered as the engine should be supplied with its own combustion air filters, and fallout particles, if present, are not expected to affect the performance of the engine in any measurable way for the duration of the emergency period. Care should be taken in protecting the above-ground inlet and discharge vents from damage by falling debris or vandalism. The ventilation air inlet and discharge openings inside the engine room should be on opposite walls to prevent short circuiting. A flexibly mounted duct should be provided between an engine-mounted radiator and the discharge opening in the engine room wall to prevent recirculation of the cooling air. The flexible duct will prevent engine vibration from being transmitted to the wall and will protect the duct from fatigue failure due to vibration.

Gasoline fuel is significantly more volatile than diesel fuel and, consequently, represents a greater fire and explosion hazard. Local codes may seriously restrict the use of gasoline fuel in some community shelters. These codes should be followed rigidly where possible, but it may be necessary to obtain permission for exceptions or compromises. Aside from the need for greater caution in handling the fuel, gasoline engine fuel system components external to the engine would be similar to diesel engine fuel system components.

The use of LPG fuel may also be in conflict with code restrictions. The fumes from LPG fuel are also highly explosive and a fire hazard. In addition, LPG fuel is stored under relatively high pressure which requires additional safety

precautions. Again, the local codes should be followed wherever possible and exceptions, where necessary, worked out in advance with local authorities.

Gasoline and LPG engine exhaust gases are toxic; consequently, the exhaust system of the spark-ignition engines will have to be installed with care to avoid any possibility of leaks. In other respects the same exhaust system requirements which apply to diesel engines also apply to spark-ignition engines.

Maintenance

Gasoline engines are not designed for as long a service life as diesel engines. On the other hand, LPG engines, because of the cleaner burning characteristics of the fuel, could be expected to have about as long a service life as the diesel engines. The industrial-type gasoline engine, even considering its shorter service life, is not expected to require any maintenance during the relatively brief emergency period. The maintenance problems which may be encountered during stand-by will be due mainly to the effects of long-term storage on the fuel system, lubricating system, cooling system, and starting system. These problems will be similar to those encountered with diesel engines and they are discussed in later sections of this report.

Safety and Human Comfort

The noise level in the engine room will be slightly lower with spark-ignition engines than with diesel engines. However, the noise level will be great enough to warrant using the same techniques for sound absorption and attenuation as were suggested for the diesel engines. Gasoline and LPG fuels, as was previously mentioned, are more dangerous than diesel fuel from the standpoint of fire and explosion hazards. Also, as previously mentioned, the spark-ignition engine exhaust gases are toxic while the diesel engine exhaust gases are not. Spark-ignition engines are provided with safety devices similar to those on diesel engines such as high water temperature and low water pressure cutoff devices. In addition, as with the diesel, spark-ignition engines can be provided with overspeed cutoff devices.

Economics

Figure 16 shows approximate purchase prices of spark-ignition engines in the 5- to 500-hp range. These data were obtained for engines supplied for electric generator service. Although spark-ignition engines are smaller and lighter than diesel engines of the same power output, the cost of installation is not expected to be significantly different. Fuel costs for spark-ignition engines will be approximately twice as great as for diesel engines. A two-week supply of gasoline for a 100-hp power source would cost about $600, and a two-week supply of LPG fuel would cost about $550.

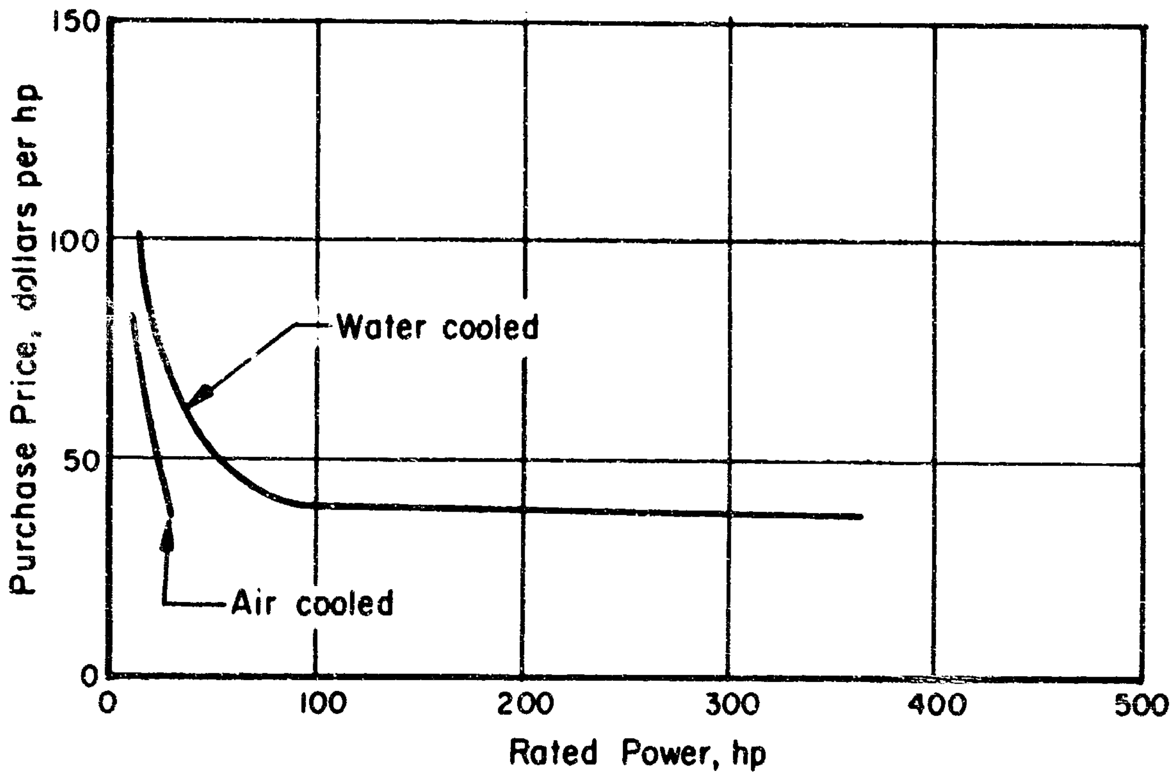


FIGURE 16. APPROXIMATE PURCHASE PRICES OF GASOLINE AND LPG ENGINES

Brief consideration was given during this study to using replacement automobile engines as prime movers because the automotive industry, with its mass production, produces engines probably for as low a manufacturing cost as could be found anywhere. Therefore, two manufacturers of gasoline engines for both industrial and passenger-car use were consulted for the over-the-counter prices and the recommended continuous-duty ratings of equivalent industrial and passenger-car engine models for auxiliary power use. However, the quoted total prices were within a few dollars of each other and the continuous-duty power ratings were within a few horsepower of each other for the same basic engine model.

Gas Turbines

The gas turbine is not a new power source technologically, but only recently have gas turbines been put to commercial use other than in aircraft. The chief advantage of the gas turbine is high power from a small package. This explains the successful aircraft application. However, basic simplicity and potential reliability are gas turbine characteristics which are playing a large role in the trend to more widespread commercial application.

Gas turbines have been made in many types and forms. but for community shelter prime mover application only single-shaft or split-shaft (free power tur'ine and regenerative or nonregenerative configurations need be considered.

Figures 17 and 18 show typical specific volume and weight data for gas turbines up to 500 shaft hp output of the type most applicable to community shelter auxiliary power systems. These data represent turbines which are now or soon will be available commercially.

Very few gas turbines are actually in production at present. Moreover, those that are available usually are intended for specific military or defense applications. Therefore, it is not very meaningful to discuss speed and size ranges available except in a general way. The turbine power shaft rotates at 30,000 to 50,000 rpm; consequently, gear reduction units are necessary to provide useable output shaft speeds. Gas turbines as small as 25 shaft hp are listed as available, and many units over 500 hp are actually in service.

Performance and Control

Figures 19 and 20 show specific fuel consumption and combustion air requirements for gas turbines. These data are considerably less reliable than the similar data for compression-ignition and spark-ignition engines because only relatively few gas turbines of each of the many available model types have been tested in actual commercial service.

Gas turbines are more sensitive to intake and exhaust conditions than either compression-ignition or spark-ignition engines. According to data from several manufacturers, an average derating for ambient temperature is 6 per cent per 10-F rise and for altitude (or intake pressure) 4 per cent per 1,000 feet.

The single-shaft gas turbine is the more responsive to speed and load changes and is the more easily governed. The split-shaft turbine, however, can be more easily started without decoupling the load, and is more adaptable to a wide variation in output speed. The control system of a gas turbine is relatively simple and is frequently automated so that the operator can merely push the start button. A governor in the fuel system controls the fuel feed according to the load demand for a given constant shaft speed.

Installation

Gas turbines are inherently vibration-free and, therefore, they can be very easily mounted with a minimum of isolation equipment. At least 2 feet of space should be provided around the turbine for convenience of servicing and maintenance. The turbine should be located convenient to the fuel supply, exhaust outlet, and ventilation air outlet.

Radiation and miscellaneous heat losses for a gas turbine will be higher in proportion to the power output than for compression- and spark-ignition engines unless the radiating surfaces are well insulated. It is more common practice to provide this insulation with gas turbines than with piston engines because of the simpler shape and smaller size of the gas turbine. Gas turbines require 5 to 8

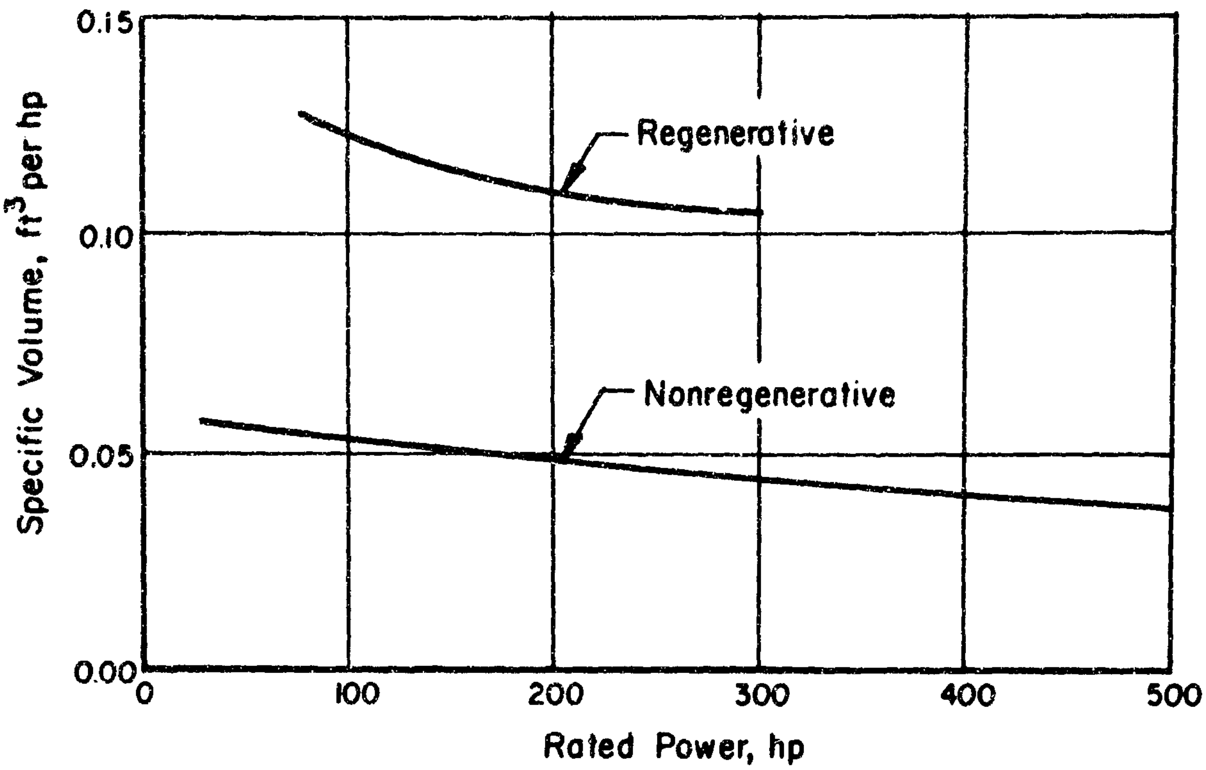


FIGURE 17. SPECIFIC VOLUME OF GAS TURBINES

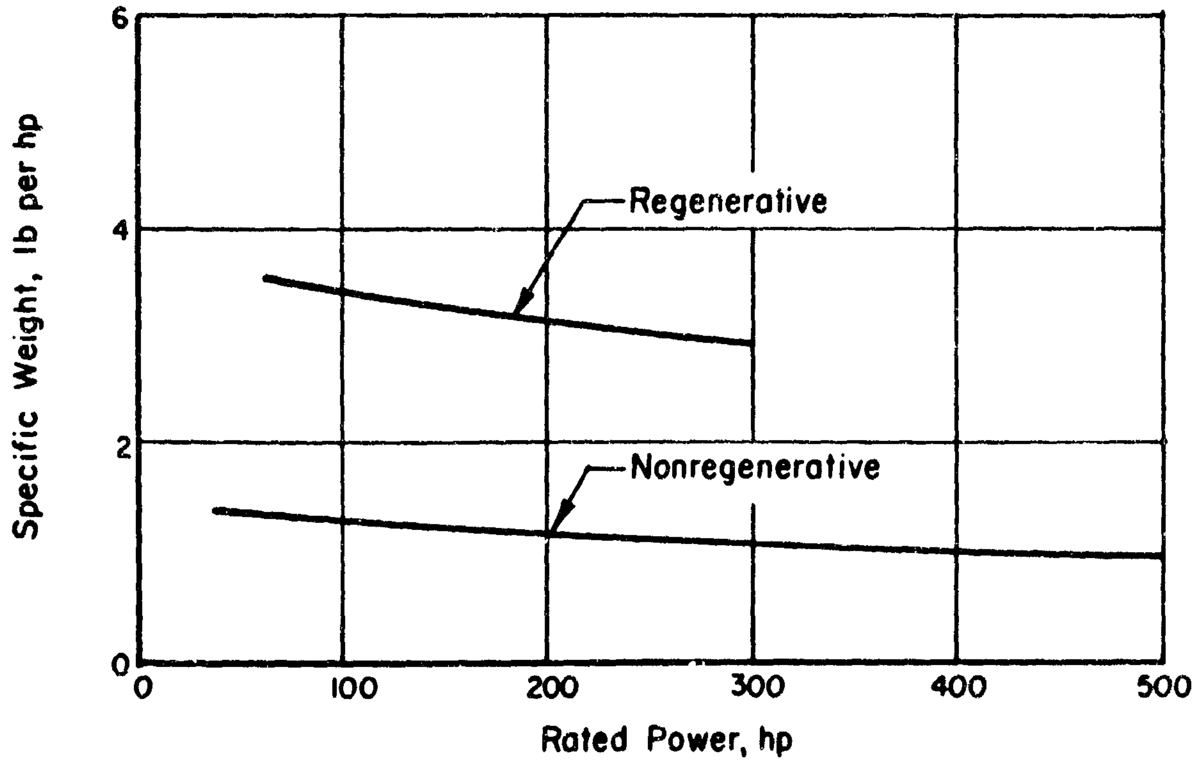


FIGURE 18. SPECIFIC WEIGHT OF GAS TURBINES

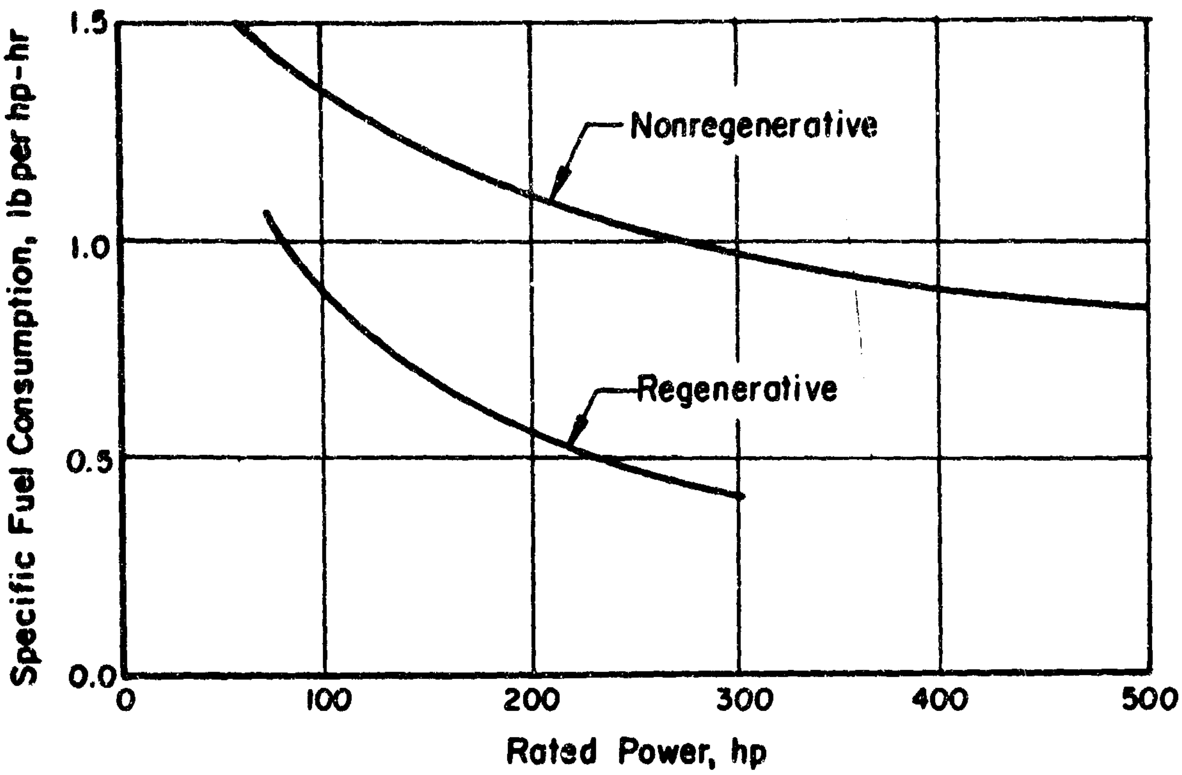


FIGURE 19. SPECIFIC FUEL CONSUMPTION OF GAS TURBINES

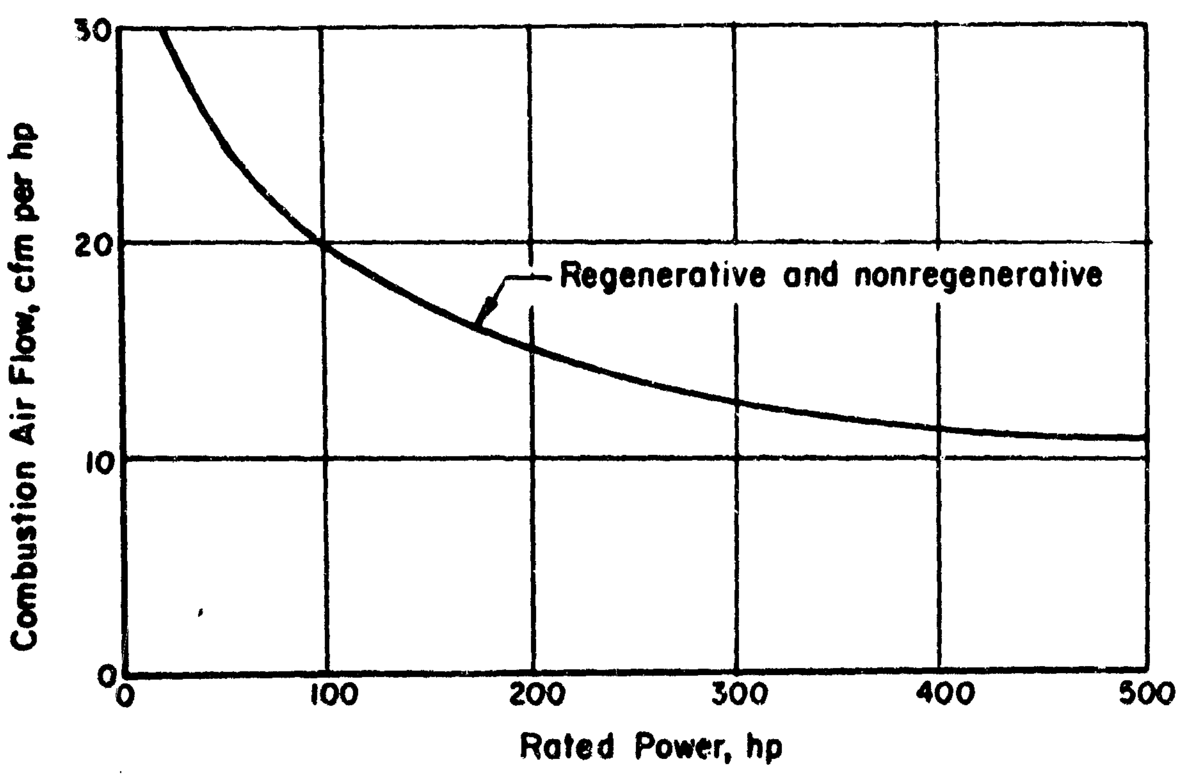


FIGURE 20. COMBUSTION AIR REQUIRED FOR GAS TURBINES

times as much combustion air as piston engines; consequently, the ventilation air requirement for a gas turbine prime mover, even with insulation, would be greater than for a piston engine prime mover unless an air-cooled radiator or condenser were used in the engine room for the piston engine.

Fuel system requirements for a gas turbine would be similar to those for compression-ignition and spark-ignition engines. Gas turbines are capable of burning any liquid or gaseous fuel. The use of leaded gasoline requires special preparation and equipment to combat lead deposits and corrosion.(1) For gaseous fuel an injection pressure of 60 to 120 psig is required. LPG fuel can probably fulfill this requirement under normal storage and supply conditions; however, if the supply pressure at the turbine is too low, a turbine-driven gas compressor can be provided.

The exhaust gases from a gas turbine are relatively nontoxic and contain 2 per cent or less of CO_2. The exhaust duct and the discharge opening for a gas turbine prime mover must be large compared with those for piston engines because of the considerably greater quantities of air handled.

Maintenance

No maintenance requirement is anticipated for a gas turbine during an emergency operating period. The stand-by maintenance requirements of gas turbines would be significantly less than those of piston engines because of the design simplicity and the lack of rubbing parts. Protection of surfaces from corrosion is still desirable but not critical. The lubrication system of a gas turbine should be prepared for storage by replacing the operational lubricant with a preservative and distributing the preservative throughout the system. Preservative should also be sprayed into the combustor and on the turbine and compressor blades.

Safety and Human Comfort

The noise level with a gas turbine should be no greater than with a piston engine if proper noise attenuation equipment is provided. The simplest approach to gas turbine noise reduction is to completely duct the inlet and exhaust flows, to provide at least one elbow to each duct, to thoroughly insulate the ducts, and to direct the duct openings vertically. The cost of silencing a gas turbine would be slightly greater than with piston engines because the ducting is generally fairly large and must be practically custom-built for each installation.

Gas turbines are provided with safety shutdown devices for overspeed, overtemperature, low oil pressure, and high oil temperature. The usual hazards associated with the different fuels applies for gas turbines as for piston engines.

Economics

Figure 21 shows approximate purchase prices of gas turbines in the 5- to 500-hp range. These data were obtained directly from manufacturers and reflect present low-volume production. Some manufacturers predicted future high-volume production and, consequently, future purchase prices of about a half or a third of the prices given in Figure 21. Installation costs of gas turbines would be very small with the exception of the cost of silencing. A two-week supply of fuel (assuming No. 2 diesel oil) for a 100-hp gas turbine power source would cost about $670 if a regenerative turbine was used and $1,000 if a non-regenerative turbine was used.

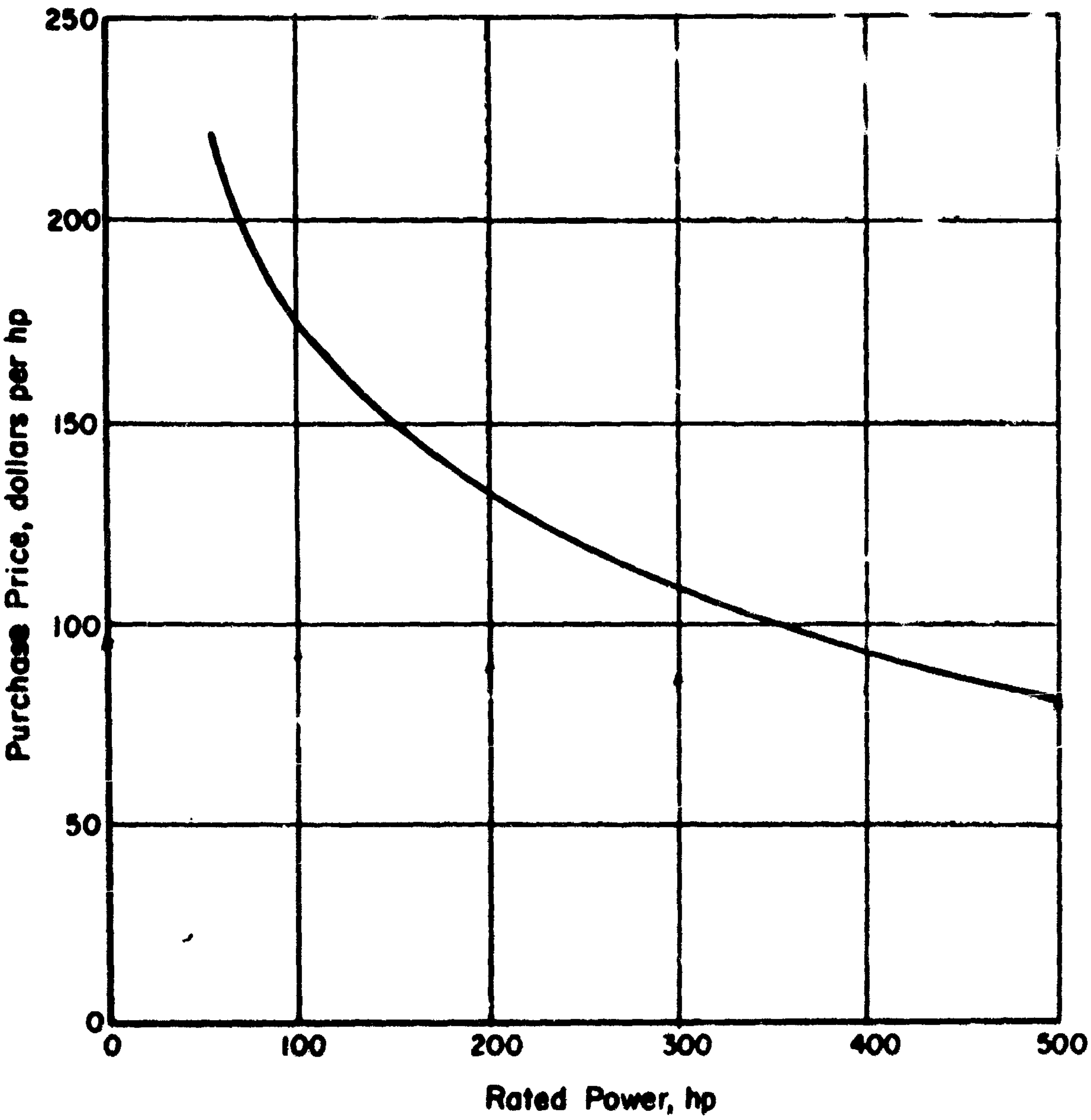


FIGURE 21. APPROXIMATE PURCHASE PRICES OF GAS TURBINES

Steam-Powered Prime Movers

The steam-powered prime mover was considered in this study because it ordinarily is included in the category of conventional power sources. However, it is not conventionally used in the relatively low power range being considered for community shelters. Complete steam power systems are most frequently found in large public utility power-generation stations. The chief advantages of steam-powered prime movers over other prime movers in this application are: the ability to burn cheap solid fuels such as coal, the lack of stringent practical size limitations, and the extremely long operating life.

Upon critical examination of these general advantages it becomes apparent that not one of them is a particular advantage with respect to community shelter application. It is significant that steam power has not been widely used except in central power stations and ship propulsion systems where large power capacity, long life, and operating economy are governing factors. However, this is not sufficient reason to rule out the steam system, therefore, an investigation of the suitability of steam power for use in community shelter auxiliary power systems was conducted. This investigation was brief however, because the results weighed heavily against the use of steam power in a community shelter.

The steam-powered prime mover was not studied in great detail. In the following discussion the steam-powered prime mover is compared directly with a diesel engine on the basis of the items considered most important in a community shelter application. The material to be covered does not lend itself to organization in the previous pattern of discussion under five major headings (e.g., performance and control).

A Representative System

A typical or representative steam power system includes several basic components: a steam generator, a steam engine, a condenser, and condensate and boiler feedwater pumps. In addition, it is sometimes necessary to provide a fairly extensive control system, a make-up water chemical-treatment system, high- and low-pressure steam and water piping, and heat-recovery equipment to maintain an acceptable over-all thermal efficiency.

The steam generator is a combined boiler, burner, and combustion chamber. This generator alone might be equal in size and weight to a piston engine and significantly larger than a gas turbine of comparable power output. The engine itself might be a piston type or turbine type, the latter being more frequently used for medium and large installations. The condenser would probably be approximately the same size as the engine, and together they might be about equal in size and weight to the steam generator.

Comparison of Steam and Diesel Systems

A steam system, in addition to being about twice as bulky and heavy as a diesel engine, would comprise at least two approximately equal-size units and would require nearly twice the shelter floor space for adequate servicing and inspection clearance. Complete steam power systems are not readily available in a very wide range of sizes, whereas diesel engines can be obtained in a great variety of power outputs.

A shelter-sized steam system would consume about 50 per cent more fuel by weight than the diesel system if the same fuel were used with each.[2,3] Thus, the fuel cost would be about 50 per cent higher. Coal as a fuel for the steam system would appear to be attractive from the standpoint of economy and resistance to age deterioration. Also, coal would actually require very little more storage space than fuel oil because of its high density. However, the equipment necessary for automatic and reliable handling of the coal would be extremely expensive. The steam system would require more combustion air than the diesel engine to match the higher fuel consumption. More cooling water (or air) for the condenser would also be required with the steam system.

The steam system would be less sensitive to ambient temperature and pressure conditions than the diesel engine, because the combustion air would be supplied by a blower which could be oversized without a very significant cost penalty. The steam system is superior to the diesel engine with respect to noise and vibration. A steam power system will not contribute significantly to the noise level in the shelter; hence, it will require no acoustical treatment of the engine room.

The control requirements with steam systems are considerably more critical than those with diesel engines because of the explosion hazard. A partial listing of necessary controls includes: automatic combustion controls, feed-water regulator, draft regulator, steam pressure regulator, turbine steam flow, and speed control. A steam power system is less adaptable to completely automatic operation, and more experienced persons are required to service and operate steam systems than diesel engines. The safety devices usually provided include: low-water fuel cutoff, overpressure cutoff, safety and blowdown valves, and flame-out protection.

Installation of a steam system would be more costly and complex than installation of a diesel engine because of the separate units and the piping which is generally custom-built at the site. The same or similar maintenance problems would be faced with a steam system as with a diesel engine. The purchase price of a steam system would probably be greater than twice that of a diesel engine, particularly in the very small sizes.

The steam power system presents so few advantages for community shelter application and so many serious disadvantages that it does not warrant being considered any further in this report. Therefore, the following discussions of auxiliary components, systems, and requirements for the total auxiliary power system will be based on the assumption that either a compression-ignition engine, a spark-ignition engine, or a gas turbine would be used as the prime mover.

STARTING SYSTEMS

All of the prime movers considered practical for shelters require some type of externally powered starting device. This external power may be in the form of manual energy or stored electric, pneumatic or hydraulic energy, or stored chemical energy as in the case of a pilot engine. The manual energy can be used directly to crank the engine or it can be used in a stored energy system. Although manual stored energy systems are not commercially available, they are attractive for shelter use from the standpoints of cost, maintenance, and reliability. Electric, pneumatic, and hydraulic systems are commercially available for the types and size range of prime movers considered practical. Pilot engines are available for only a limited range of larger prime mover sizes and types. Because of this they were not considered further in this study.

Electric Starting

Figure 22 is a schematic illustration of an electric starting system.

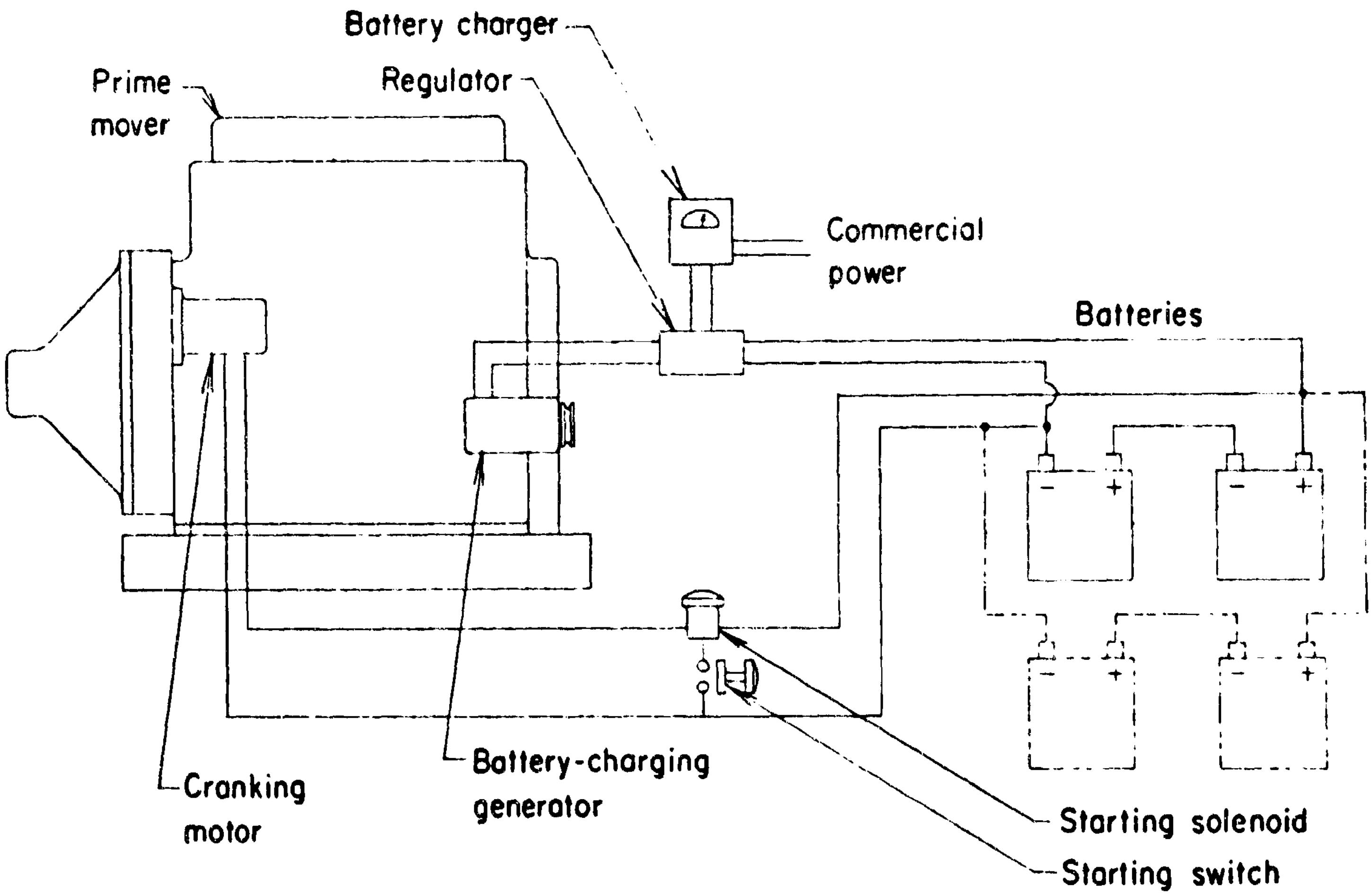


FIGURE 22. ELECTRIC STARTING SYSTEM

The major components of this system are: (1) batteries, (2) starting switch and solenoid, (3) cranking motor, (4) battery charger, and (5) regulator. All of this equipment is readily available commercially for either 12- or 24-volt systems. The 12-volt systems would be used with the smaller prime movers and the 24-volt systems with the larger.

The batteries are the most critical component of an electric starting system. In fact, the primary maintenance requirement is to maintain the batteries at or near full charge at all times. The illustration shows two alternative means for maintaining battery charge: (1) commercial-power battery charger and (2) engine-driven charging generator. A system may require one or the other of these methods or both together. The engine-driven charging generator will be useful if frequent periodic exercising of the power source is planned. The batteries could probably be maintained at full charge by this means if the prime mover were exercised at least once a month for a period of not less than 2 hours each time. If less frequent exercising is planned, or if a greater reliability of starting is essential, a commercial-power battery charger should be included in the starting system along with the engine-driven charging generator.

Four principal types of commercial-power battery chargers are available: (1) trickle charger, (2) variable-rate charger, (3) two-rate charger, and (4) high-rate charger. The trickle charger supplies a low charging current at a constant-voltage potential. Consequently, the battery cannot be overcharged under normal conditions. However, a considerable length of time is required for charging a discharged battery.

With the variable-rate charger the charging cycle begins at a high rate, depending on the battery potential. The rate tapers off as the battery becomes charged until a constant rate equivalent to that of the trickle charger is reached. With the two-rate charger, the charging cycle also begins at a high rate. After a certain battery potential has been reached the charger switches to a low rate. The high-rate charger charges at a high rate for a short predetermined interval. It is then switched off automatically and remains off until the specific gravity of the battery electrolyte falls below a minimum safe value. Then the charging cycle is again triggered.

The battery most commonly used in emergency standby power systems is the lead-acid wet-cell battery. This battery has a potential life of 3 to 15 years, depending on the quality of manufacture and materials and on the means of maintaining the charge. Although it is very bulky, it is the least expensive of the commercially available batteries capable of providing the high rates of discharge required for starting large diesel engines. A sealed lead-acid wet-cell battery has recently become commercially available. Being sealed it would require significantly less standby maintenance than the unsealed type. However, it is presently limited to very low rates of discharge. Lead-acid wet-cell batteries are also available as dry charge batteries, i.e., the electrolyte is stored in a separate container and added when the battery is put in service. Dry charge batteries have a shelf life of about one year in the dry state and require a charging period after the electrolyte is added to ensure that they will deliver rated power. All lead-acid wet-cell batteries lose capacity as the battery temperature decreases. At 0 F the capacity is reduced by 50 per cent from its rated capacity at 80 F.[4]

Other batteries considered in this study are: nickel-iron, nickel-cadmium, silver-zinc, and silver-cadmium. The nickel-iron (Edison) cells are very long lived batteries, up to 15 years or more, and are extremely reliable. However, to get the high rates of discharge required for engine starting a particularly large battery is required. The nickel-cadmium battery is generally considered to be more reliable than the lead-acid battery and to require less maintenance. The estimated life of the nickel-cadmium battery is 15 to 25 years and the cost is about four times that of the lead-acid battery. The nickel-cadmium battery is capable of producing half its full-rated capacity at about -30 F.[4] The sealed nickel-cadmium battery is in the same category as the sealed lead-acid battery. That is, it is significantly more reliable and requires much less maintenance than the unsealed nickel-cadmium battery but it is not capable of delivering the high rate of discharge required for engine starting. Silver-zinc and silver-cadmium batteries are very compact, but also very expensive and short lived. The silver-zinc battery has an estimated life of about one year; the silver-cadmium battery has an estimated life of about 5 years. A reserve-type silver-zinc battery is also available; it can be stored a great many years in the dry state. Electrolyte is added when the cell is needed. The cell is discharged during use and is not rechargeable. At the present time, the unsealed lead-acid and nickel-cadmium batteries are the most logical choices for providing power for electric starting systems.

Standby maintenance of electric starting systems involves maintaining a charge in the batteries, inspection of components, and periodic exercising through a starting cycle. Battery charging, the most critical and demanding maintenance requirement, has already been discussed in some detail. Batteries give off small quantities of explosive and toxic gases while being charged; hence, the power system enclosure should be periodically checked for dangerous concentrations of these gases or the batteries should be vented to the atmosphere. The battery condition can be evaluated by measuring the cell voltages and the specific gravity of the electrolyte, and by checking for corrosion at the terminals, for liquid leaks, and the electrolyte level. A record should be kept of battery condition so that any tendency toward deterioration can be discovered in adequate time for servicing or replacement.

The other major components of the electric starting system: starting switches and solenoid, cranking motor, and voltage regulator, can be expected to tolerate long idle periods without deterioration. This tolerance can be improved by the use of moisture-resistant materials or by maintaining a reasonably dry environment.

The installed costs of minimum electric starting systems vary from about \$100 up to about \$600, depending on the size and type of prime mover and on the complexity of the system. Diesel engines require slightly more expenisve starting systems than gasolien and LPG engines because of the higher cranking energies required. Gas turbine starting systems are slightly less expensive than gasoline and LPG engine starting systems. Greater reliability of starting can be obtained at greater cost with more complex battery charging equipment and with a greater number of batteries in the system.

Hydraulic Starting

Figure 23 is a schematic illustration of a hydraulic starting system.

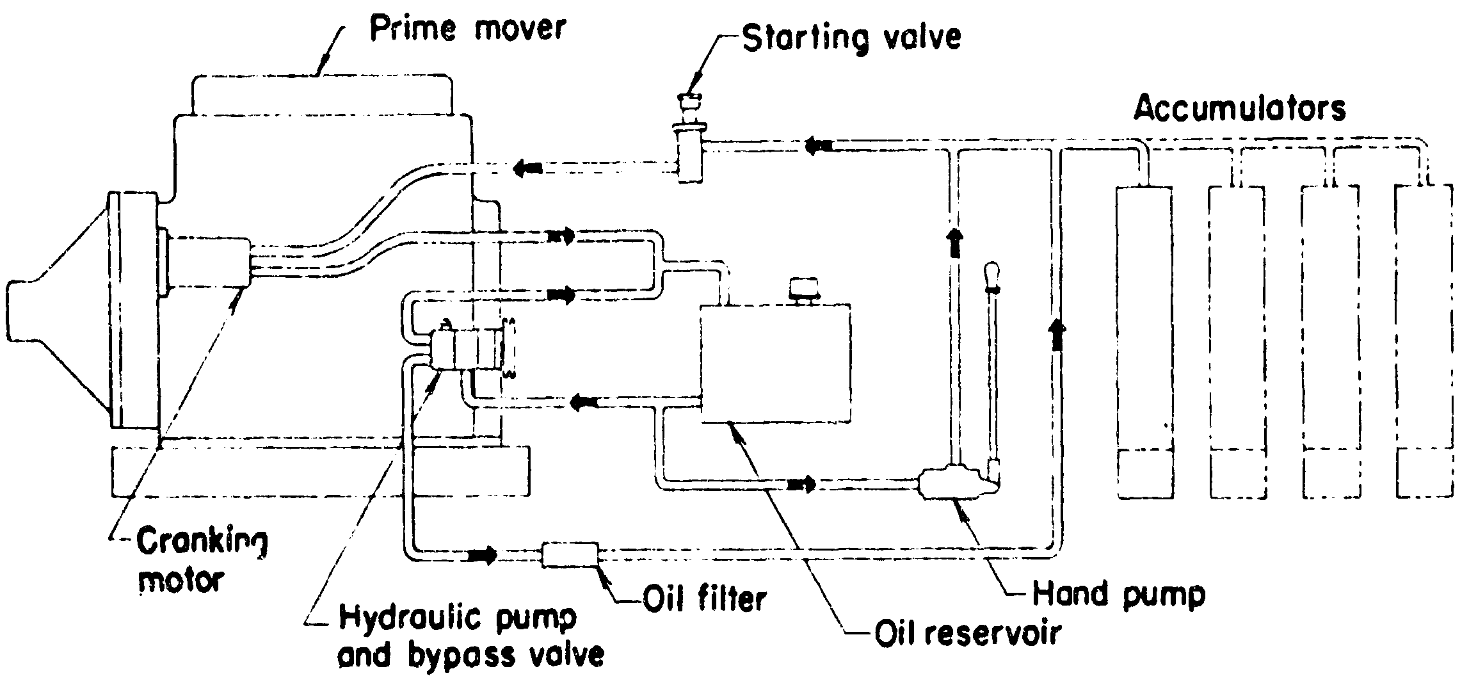


FIGURE 23. HYDRAULIC STARTING SYSTEM

The basic components of this system are: (1) cranking motor, (2) accumulators, (3) filters, (4) reservoir, (5) charging pump, (6) hand pump, and (7) hand starting valve. Hydraulic starting systems are commercially available for all types of prime movers except gas turbines.

The maintenance of a hydraulic starting system involves merely checking the accumulator pressure and exercising through a starting cycle. An engine-driven charging pump, as shown in the illustration, is desirable if frequent exercising of the prime mover is planned. However, a separate, electrically driven charging pump is more flexible and probably slightly more reliable in maintaining the system pressure during standby. The working pressure for hydraulic starting systems is usually between 1000 and 3000 psi.

Hydraulic starting systems provide very high cranking speeds, but for only very short times. The reserve capacity of hydraulic starting systems is quite limited; a large number of accumulators are required for the same number of starting attempts that could be supplied by just a few batteries. However, a compensating feature is that the system can be recharged readily by hand. Consequently, the necessity for a large capacity accumulator is eliminated, and the number of starting attempts is not limited by the equipment.

Hydraulic starting system components are not greatly affected by temperature, moisture, and dirt. Therefore, they are easy to maintain. The life of the system is estimated to be equal to or greater than the life of the other components of the auxiliary power system. Hydraulic starting system components are generally more expensive than components of electric starting systems. Minimum hydraulic starting systems range in cost from $400 to $2,000, depending on the size and type of power source.

Pneumatic Starting

Figure 24 is a schematic illustration of a pneumatic starting system.

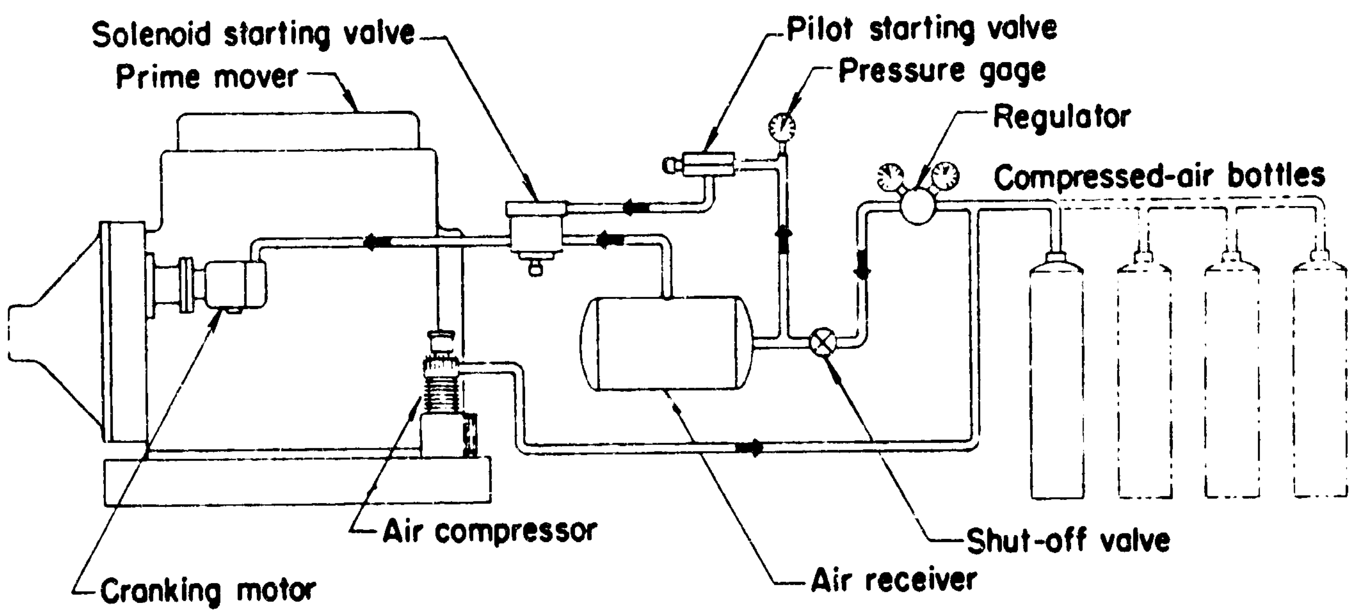


FIGURE 24. PNEUMATIC STARTING SYSTEM

The basic components of this system are: (1) cranking motor, (2) starting valves, (3) air receiver, (4) air tanks, and (5) air compressor. Pneumatic starting systems are available for nearly all types and sizes of power sources, except gas turbines. The pneumatic starting system is similar to the hydraulic starting system in providing a high cranking speed for a short cranking period. The working pressure for a pneumatic starting system is usually between 100 and 150 psi.

Standby maintenance for pneumatic starting systems involves chiefly visual checking of system pressure and periodic operation through a starting cycle. The life of the components of pneumatic starting systems will be equal to or greater than the life of the other major components of the auxiliary power systems.

A standard-size air receiver tank for a specific prime mover would provide only about 30 seconds of starting time. Most power sources operating regularly and frequently should start within this time. However, a standby power source not regularly exercised may require significantly more starting time. Consequently, a source of reserve starting power must be provided in a pneumatic

starting system. A typical reserve power source is illustrated in Figure 24 which shows standard commercial compressed-air bottles. The 210 cubic feet of air stored at 2000 psi in each bottle would increase the normal starting period by 30 to 90 seconds. Alternatively, the air receiver could be increased in size to provide additional reserve starting power. An engine-driven air compressor, as illustrated in Figure 24, would be satisfactory for maintaining a pressure in the air receiver during standby if frequent exercising of the prime mover is planned. A separate, electrically driven air compressor would be more flexible in its adaptability to any standby maintenance program, but would also be slightly more expensive.

The estimated cost of pneumatic starting systems varies between $400 and $1,000, depending on the size and type of the prime mover, the complexity of the system, and the amount of reserve compressed air provided.

Manual Starting

Manual starting systems can be divided into two categories: (1) direct such as a hand crank or rope which are commercially available and (2) stored energy such as a flywheel or falling weight which are not commercially available. Because of the limitation on human power capacity, application of the direct type is limited to small engines. The limitation on human power capacity does not apply to the stored energy type, and therefore, this type is applicable to much larger engines.

A manual starting system must be designed such that the input requirement is within the capability of an average man. In an article by E. S. Krendel[5] test results are given for one- and two-handed cranking effort for short periods and for bicycle-type peddling for long periods. The data indicate that a young man can sustain about 0.6 hp for 30 seconds with two-handed cranking, while the upper limit of human energy output is approximately 1 hp sustained for 5 minutes by a trained racing cyclist. A strong, but untrained cyclist can maintain about 0.6 hp for the same length of time. The most effective crank radius is evidently the largest which can easily be spanned, and this appears to be about 1 foot. A typical short-term cranking speed is about 80 rpm, although speeds up to 260 rpm were recorded at low loads. It appears that a reasonable standard of human output for the purposes of this study would be two-handed cranking at 80 rpm with 27 pounds average tangential force applied at a 12-inch radius. The power developed under these conditions is about 0.41 hp, and it should be sustainable for between 1 and 2 minutes. If larger outputs are required, tandem cranking or even tandem peddling could be adopted. The use of more than two men would probably get into the area of diminishing returns because of increased complications and difficulty with coordination of effort.

In addition to knowledge of human power capability it is also necessary to know engine motoring torque requirements. Unfortunately, low-speed motoring torque data for engines are difficult to obtain, and nearly impossible to calculate because of the large number of variables involved such as oil viscosity, condition of piston rings, bearing clearances, accessory drives, etc. Information for this study was obtained by taking manufacturer's maximum torque ratings for a typical air starter and multiplying by a typical 12-1 pinion-to-flywheel ratio to get an

estimated maximum starting torque at the crank shaft. This approach should yield fairly conservative values. A maximum starting torque of 348 ft-lbs was estimated to be required by a typical 100-hp diesel engine which was selected for the calculations of over-all system size and component requirements for the energy-storage starting systems.

Manual Direct Starting

Manual direct starting systems are inexpensive, require almost no maintenance, and are as reliable as the manpower available to operate them. Rope starters are available for gasoline engines up to 10 or 12 hp and for diesel engines up to about 5 hp. Hand-crank starting is not practical for gasoline and LPG engines over 40 hp and for diesel engines over 20 hp. Gas turbines of 50 to 75 hp can be hand-crank started if a means is provided to unload the compressor during the cranking period. A hand crank could be provided as an emergency starting means even for prime mover sizes that are normally considered too large for hand starting, this would provide a last resort possibility in the event the main starting system failed to start the engine. A hand-crank starting system would have to be suitably constructed to be of any real value. For example, the crank should be supported in good quality bearings and should have a long and comfortable handle.

Manual Energy-Storage Starting

Three manual energy-storage starting methods were investigated during this study: (1) flywheel, (2) falling weight, and (3) spring. None of these were found to be commercially available in any form, but all are considered feasible. All the systems offer high reliability, low maintenance requirements, and reasonably low first cost. The flywheel system would probably require the least amount of floor space. The spring and falling weight systems would be more efficient. The falling-weight system would have the shortest activation time or require the least time to develop enough energy for a single starting attempt. However, the activation time for all three systems probably would not exceed 2 or 3 minutes.

Flywheel-Type Energy-Storage Starting. Figure 25 is a schematic illustration of a flywheel-type energy-storage starting system. The system is shown coupled to the engine through a conventional starter pinion assembly which provides a 12-1 gear ratio in itself.

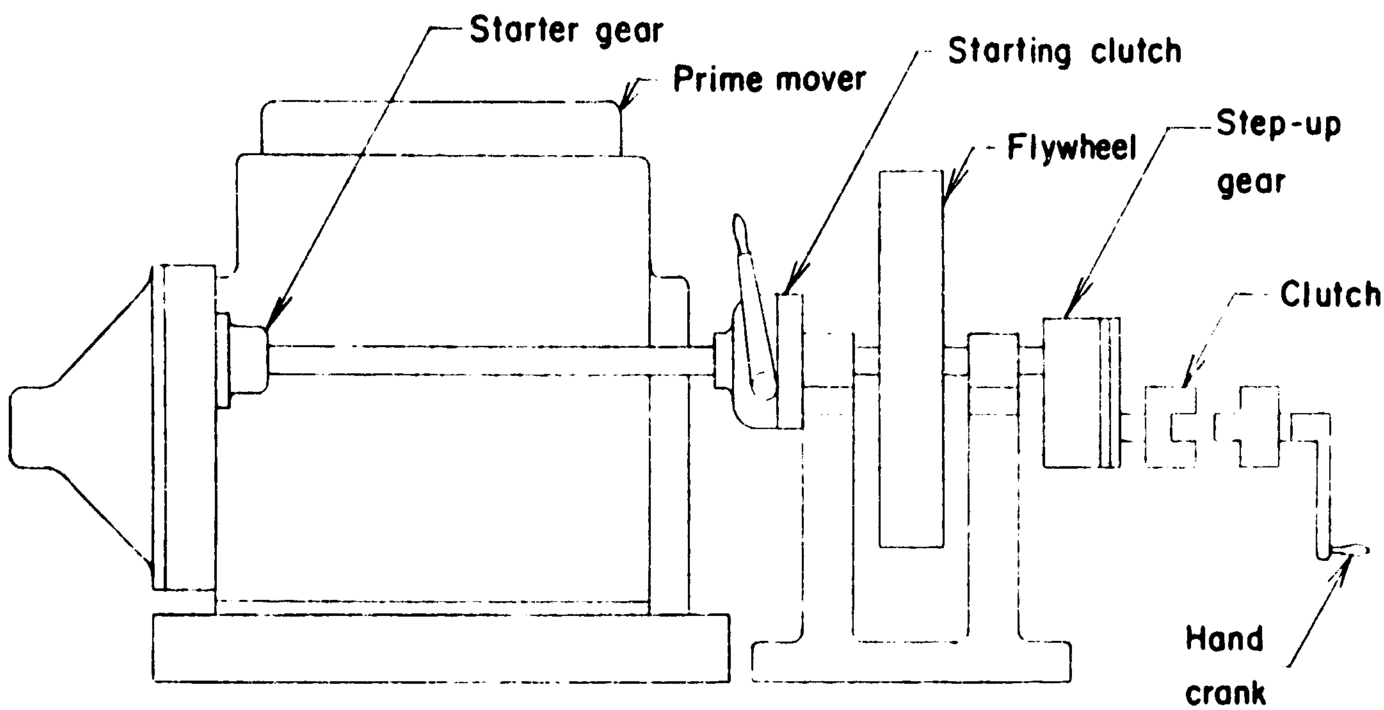


FIGURE 25. FLYWHEEL-TYPE ENERGY-STORAGE STARTING SYSTEM

To initiate the starting cycle, the operator disengages the starter clutch between the flywheel and engine and engages the clutch between the crank and flywheel. He then cranks the flywheel through the step-up gears until sufficient kinetic energy is stored in it. Finally he disengages the crank (for safety) and engages the starting clutch. The initial shock of momentum transfer is absorbed by the slipping clutch, which is set to allow slippage at a torque slightly above that required to get the engine past the first compression stroke.

Turning an engine over 4 or 5 revolutions should be more than sufficient to start it. At the estimated maximum resisting torque of 348 ft-lbs, for the 100 hp diesel being used as an example this would require 11,000 ft-lbs of energy. To allow for acceleration of the engine parts, the energy requirement should be increased by a factor of 1.5. For the example, the energy requirement would be 16,500 ft-lbs. Flywheels of inertia starters used on World War II aircraft have been satisfactorily run between 10,000 and 14,000 rpm; consequently a speed of 10,000 rpm was arbitrarily selected for the high-speed flywheel starting system. A steel flywheel 8.2 inches in diameter and 1 inch thick (moment of inertia equals 0.363 in-lbs^2) would have the required energy at that speed. Step-up gears of 125-1 ratio would allow 80 rpm at the crank to produce 10,000 rpm at the flywheel. This is consistent with previous practice in inertia starter design. A 100-1 total step-down ratio from flywheel to engine would give 50 rpm at the crankshaft when the flywheel had slowed to 5,000 rpm, provided the clutch had stopped slipping.

The friction losses in the two gear trains would be considerable, but would be partially compensated for by the kinetic energy stored in the gears themselves. Assuming a constant tangential force of 27 lbs on a crank having a radius of 12 in. (as previously specified), the time required to get the flywheel up to speed would be about 2.5 minutes. This is twice as long as would be required on a basis of continuous 0.41 hp output, because the cranking speed must start from 0 and terminate at 80 rpm, giving an average cranking speed of only 40 rpm.

It is evident that the use of a high-speed flywheel, although consistent with established practice, is disadvantageous because of the two sets of high-ratio gearing required. Previous inertia starters were designed for high speed primarily to save weight in aircraft installations. In a community shelter, weight limitation is not critical, and it is worth while to investigate the possibility of using a larger low-speed flywheel.

If the maximum flywheel speed is limited to 1200 rpm, the flywheel shaft can be directly coupled at the point where the starter pinion normally enters the engine. This will eliminate the step-down gear set and will give a 100-rpm nominal initial starting speed at the crankshaft, assuming that a typical 12-1 pinion-to-ring gear ratio is used. A steel flywheel 22 in. in diameter and 1.5 in. thick, weighing 161 lbs will store the required 16,500 ft-lbs of energy at 1,200 rpm. The step-up ratio required for hand cranking is 15-1 in this case and allows the use of fewer and less precise gears than the 125-1 set needed for the high-speed flywheel. The time required to accelerate the flywheel remains about 2.5 minutes. Other features of the low-speed flywheel system, such as the starter clutch and the hand crank clutch would remain relatively unchanged.

Falling-Weight-Type Energy Storage Starting. Figure 26 is a schematic illustration of a falling-weight-type, energy-storage starting system. To initiate the starting cycle the operator raises the weight along the guides by means of a winch and then holds the weight by means of a winch brake. The engine cable, which then hangs loosely, is wound around the starting drum on the engine shaft by turning the drum backward against its ratchets until the cable is tight. For starting the operator disengages the crank handle and releases the winch brake. The cable end may be designed to pull free of the starting drum after unwinding, or it may be left attached. In this case, the ratchet pawls could be designed to retract by centrifigual force to avoid continuous clicking of the ratchet while the engine runs. The shock absorber would cushion the impact of the falling weight.

Because of the space limitations in community shelters, it is advisable to have the weight fall through a relatively short distance. For the 100 hp diesel engine example, a 2,750-lb weight falling through 6 feet was selected. This will product the 16,500 ft-lbs required to start the engine. With a 4.6-in. drum diameter, 5 full engine revolutions are obtained at 517 ft-lbs of static torque. This torque is about 50 per cent in excess of the estimated maximum required, hence providing adequate cranking speed for a minimum falling distance. With a winch having a gear ratio of 23-1, a 12-in. handle, and a 4-in. diameter drum, a force of 20 lbs at the crankhandle will balance the 2,750 lb weight. A cranking force of 27 lbs will raise the weight to the required 6 feet in slightly less than 2 minutes.

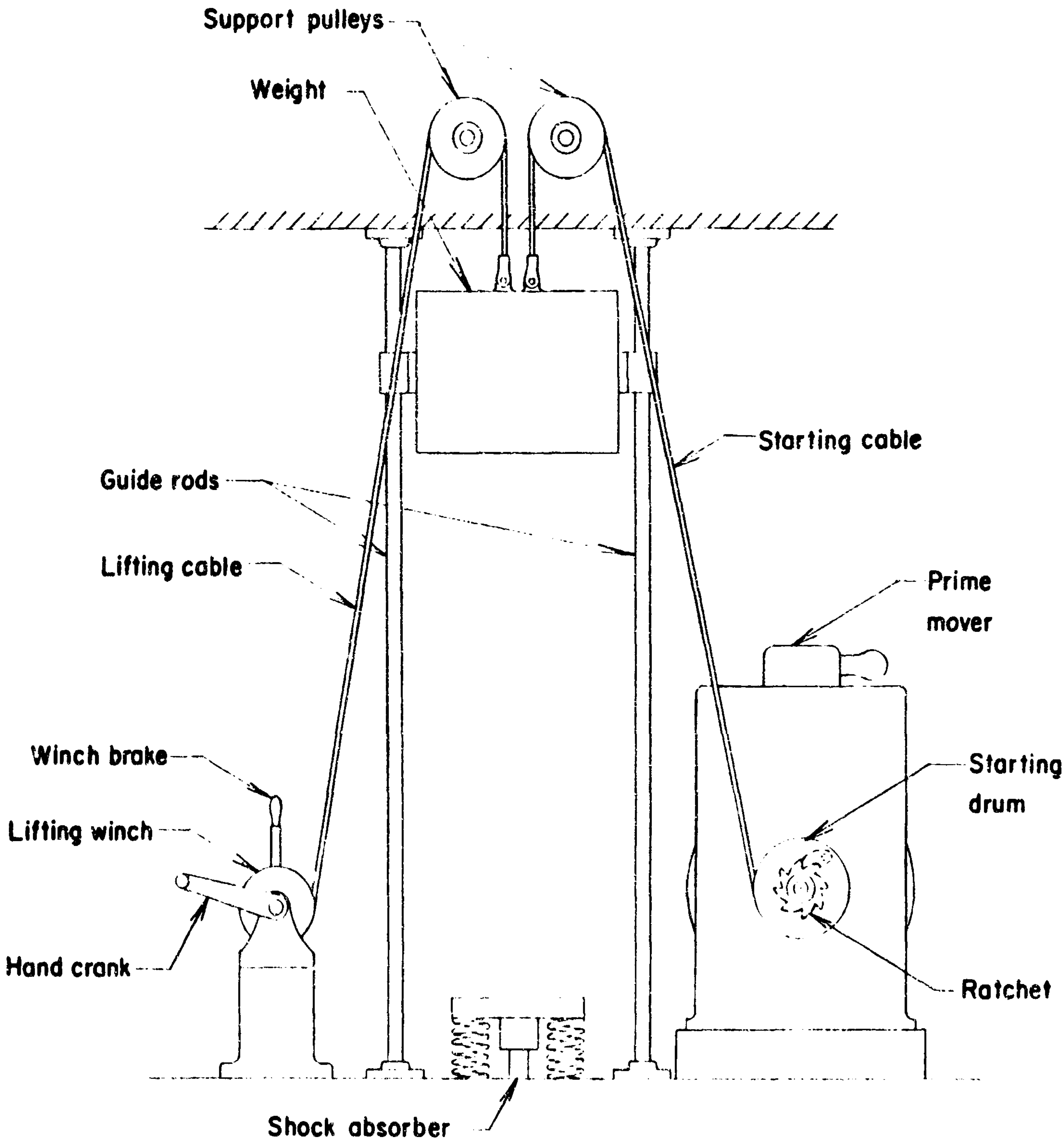


FIGURE 26. FALLING WEIGHT-TYPE ENERGY STORAGE STARTING SYSTEM

Spring-Type Energy-Storage Starting. Figure 27 is a schematic illustration of a spring-type energy-storage starting system. To initiate the starting cycle, the operator sets the brake on the starting drum and extends the springs by cranking the winch. He then sets the winch against its ratchet. For starting, the starting-drum brake is released. As with the falling-weight system, the starting drum must contain a ratchet to allow winding of the slack cable and to prevent rewinding upon starting. The cable may either pull free of the drum or remain attached to the cable drum while its ratchet idles.

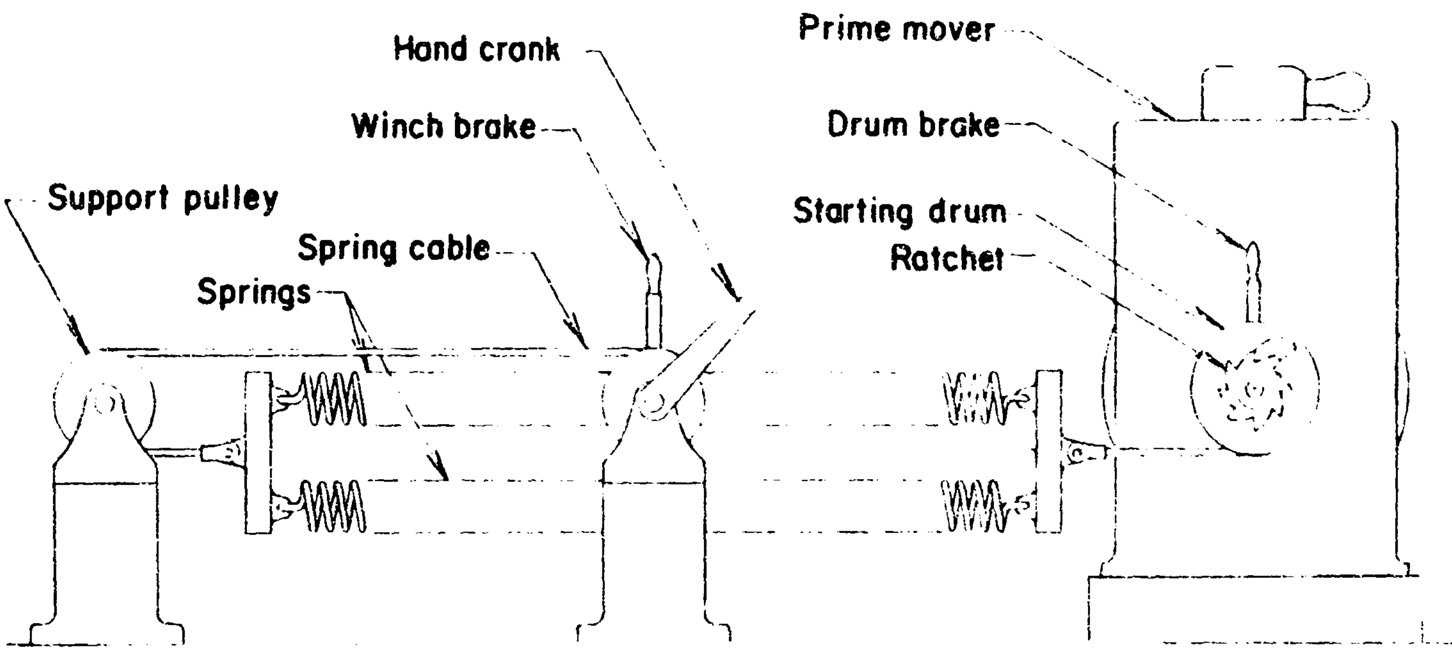


FIGURE 27. SPRING-TYPE ENERGY-STORAGE STARTING SYSTEM

A typical setup for the 100-hp diesel engine might employ two tandem coil springs of 100-in. closed length, 4-in. mean diameter, and 0.704-in. wire diameter. A spring deflection of 6 ft would provide the required 16,500 ft-lbs of energy. A 4.6-in. cable drum on the engine would allow 5 full revolutions to be made.

Unfortunately, the spring system is not inherently a constant-torque device, as is the falling-weight system. However, the initial torque of 1,050 ft-lbs is three times the estimated maximum resisting torque, and this should provide adequate cranking speed during the full 5 crandshaft turns. The average torque applied during the cycle will be 1.5 times the conservatively estimated resisting torque of 348 ft-lbs. If the same winch handle and drum dimensions are used as in the falling-weight system, but with the gear ratio doubled to 46-1, the springs could be fully extended in about 3.3 minutes, assuming 80 rpm cranking speed and a maximum force of 20 lbs at the crank handle.

It should be emphasized that all the manual energy-storage starting systems illustrated in this report are merely conceptual. Time did not permit a thorough investigation in this area. It is almost certain that many other configurations could be devised to fulfill the requirements, some perhaps significantly better than those illustrated. The primary purpose of this study was to determine the general feasibility of manual energy-storage starting systems.

Estimated Starting Systems Costs

Figures 28 and 29 show the approximate costs of starting systems for conventional power sources. Figure 28 shows cost data for starting systems for diesel engines and gas turbines and Figure 29 shows cost data for starting systems for gasoline and LPG engines. These data were obtained from manufacturers' literature and include estimates of the costs of installation and of suitable auxiliary equipment. In all cases, the installed cost of the electric starting system is lower by a significant margin. However, if standby maintenance costs were included the relative costs may be altered considerably.

It has been previously mentioned that manual energy-storage starting systems are not presently available in any form for the types of power sources considered in this study. However, to obtain some idea of the potential cost of a typical energy-storage starting system for comparison with the costs of the commercially available starting systems, an estimate was made on the basis of the low-speed flywheel-type system. Standard, readily available components were considered wherever possible in this estimate and no allowance was made for volume production or quantity buying. The estimated cost figure is \$325 for a system capable of starting a 100-hp diesel engine.

This \$325 figure has been included in Figures 28 and 29 and appears to compare quite favorably with the costs of electric starting systems. To include these data in Figure 29 the assumption was made that a system capable of starting a 100-hp diesel engine would be capable of starting a 150-hp gasoline or LPG engine. It can be assumed that a reasonable amount of design effort, volume production, and quantity buying would be reflected in a significantly lower final cost for low-speed flywheel-type energy-storage starting systems.

Starting Aids

The demands on any starting system can be significantly reduced by following certain procedures to ensure that the engine will be as easy to start as possible. The most important procedure is to make certain that the fuel and ignition systems are functioning properly. Engine manufacturers' instructions for start-up should be followed as closely as practicable. In addition, there are a number of starting aids available which are ordinarily recommended for cold weather starting but which would be of some assistance under any conditions. These include: (1) special starting fluids that can be automatically measured and sprayed or atomized under pressure into the intake manifold, (2) immersion heaters to keep the lubricating oil and engine coolant warm, (3) intake manifold air heaters, and (4) glow plugs for diesel engines.

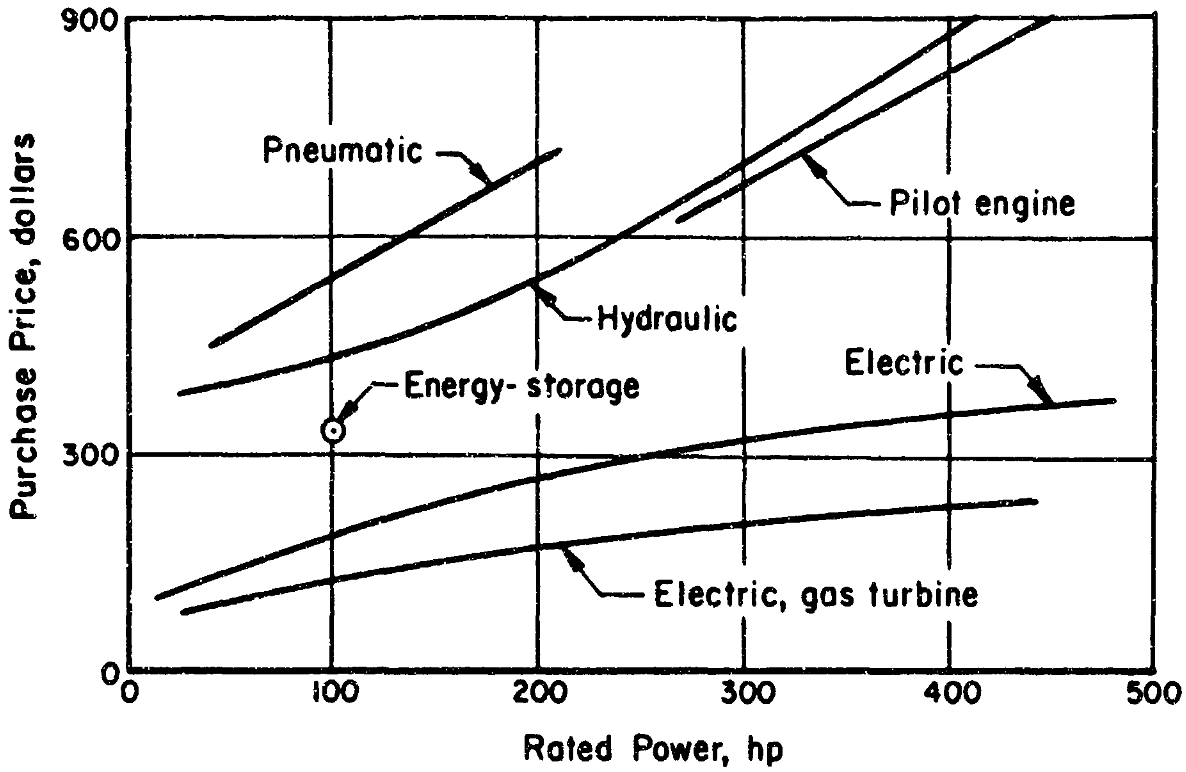


FIGURE 28. APPROXIMATE STARTING SYSTEM PRICES FOR DIESEL ENGINES AND GAS TURBINES

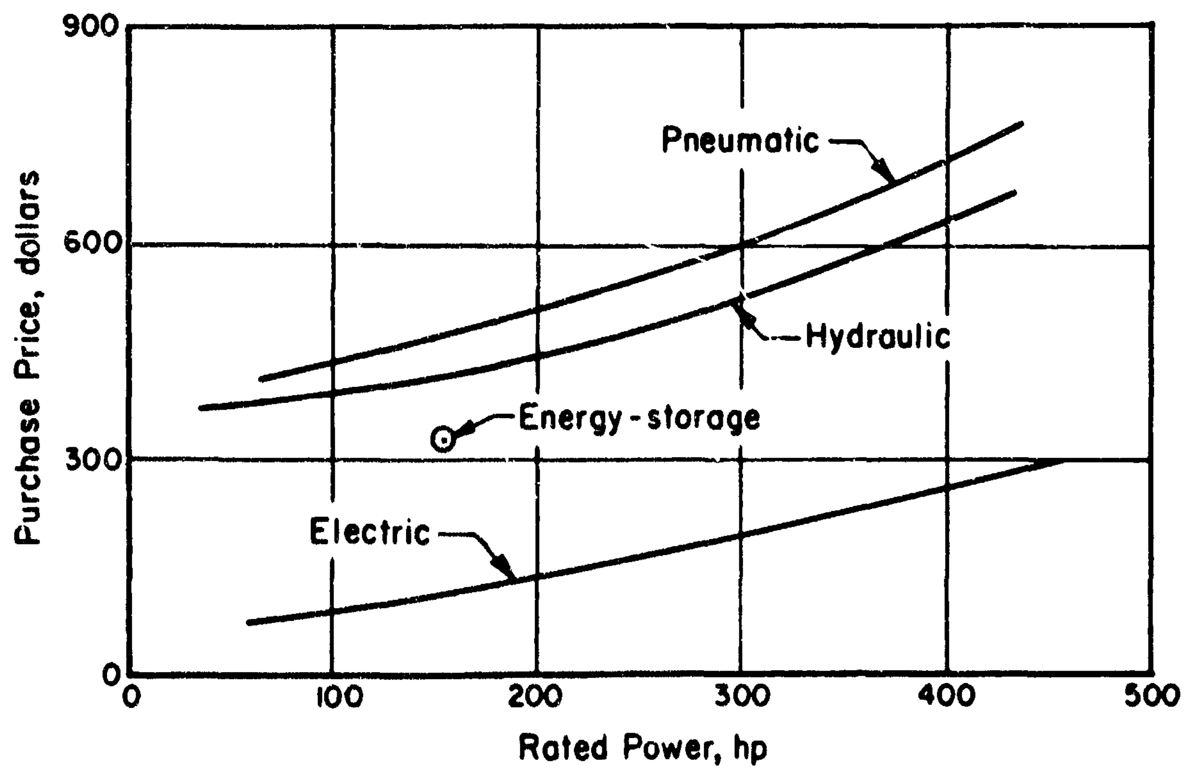


FIGURE 29. APPROXIMATE STARTING SYSTEM PRICES FOR GASOLINE AND LPG ENGINES

Most of these starting aids are relatively inexpensive and easy to use. It is estimated that total costs may range from $12 to $15 for starting-fluid spray systems and up to approximately $200 for the more elaborate hot water systems for heating the intake manifold. These costs will depend on prime mover size as well as type and complexity of equipment selected. Many of these starting aids may be obtained only from the engine manufacturer since their application can be significantly influenced by the basic engine configuration.

COOLING

Most conventional prime movers reject to a cooling medium a portion of the total energy available in fuel. Not only is there a necessity for providing a means for carrying away this rejected heat, but this same means must regulate the engine temperatures within a relatively narrow range for optimum operation. An exception to this is the gas turbine which rejects all of the fuel energy not converted to shaft power in the exhaust gases. It has been found through experience and experimentation that engine service life can be greatly extended by maintaining sufficiently high cylinder wall temperatures to prevent condensation of the various corrosive products of combustion and by maintaining lubricating oil temperatures between 200 and 280 F.(6)

The basic techniques that have been used for conventional engine cooling are air cooling and water cooling. In engines designed for air cooling, the external surface area is usually made large by finning. In such engines a mechanically driven blower or fan forces a sufficient quantity of air over the finned surfaces to carry away the rejected heat. Air cooling is generally confined to the smaller size engines and to applications in which light weight is extremely important.

In water-cooled engines, cool water is circulated through passages between an outer case and the engine cylinders. The heat absorbed by this water as it passes through the engine is ultimately rejected to the surroundings by various techniques. Radiators, heat exchangers, cooling towers, and vaporization devices are used. Nearly all industrial engines, for stationary service, are water cooled

Figure 30 shows estimated average heat-rejection rates to the coolant for several commercial prime movers. These data were obtained from manufacturers' literature and represent conditions at full load for continuous-duty type prime movers. The heat-rejection rates of these prime movers generally are consistent with the thermal efficiencies in that higher thermal efficiency results in lower cooling losses. An exception to this is the two-cycle diesel engine which has a lower heat-rejection rate than the more efficient four-cycle naturally aspirated diesel because the higher combustion-air flow rate in this type of engine results in higher heat rejection in the exhaust gases.

The data given in Figure 30 do not present the entire picture of the cooling requirements of prime movers. In general, a prime mover will reject directly to the cooling system between 20 and 35 per cent of the total fuel energy supplied, the actual quantity depending on the type of engine, the type of cooling system, the quality of manufacture, and the mode of operation. In addition, 5 to 10 per cent of the total fuel energy supplied is rejected directly to the enclosure in the form of radiation losses. Due to inefficiencies, the power transmission system will also reject heat directly to the enclosure. The total heat which must ultimately be absorbed by a heat sink or heat sinks can be as much as 50 per cent greater than shown in Figure 30.

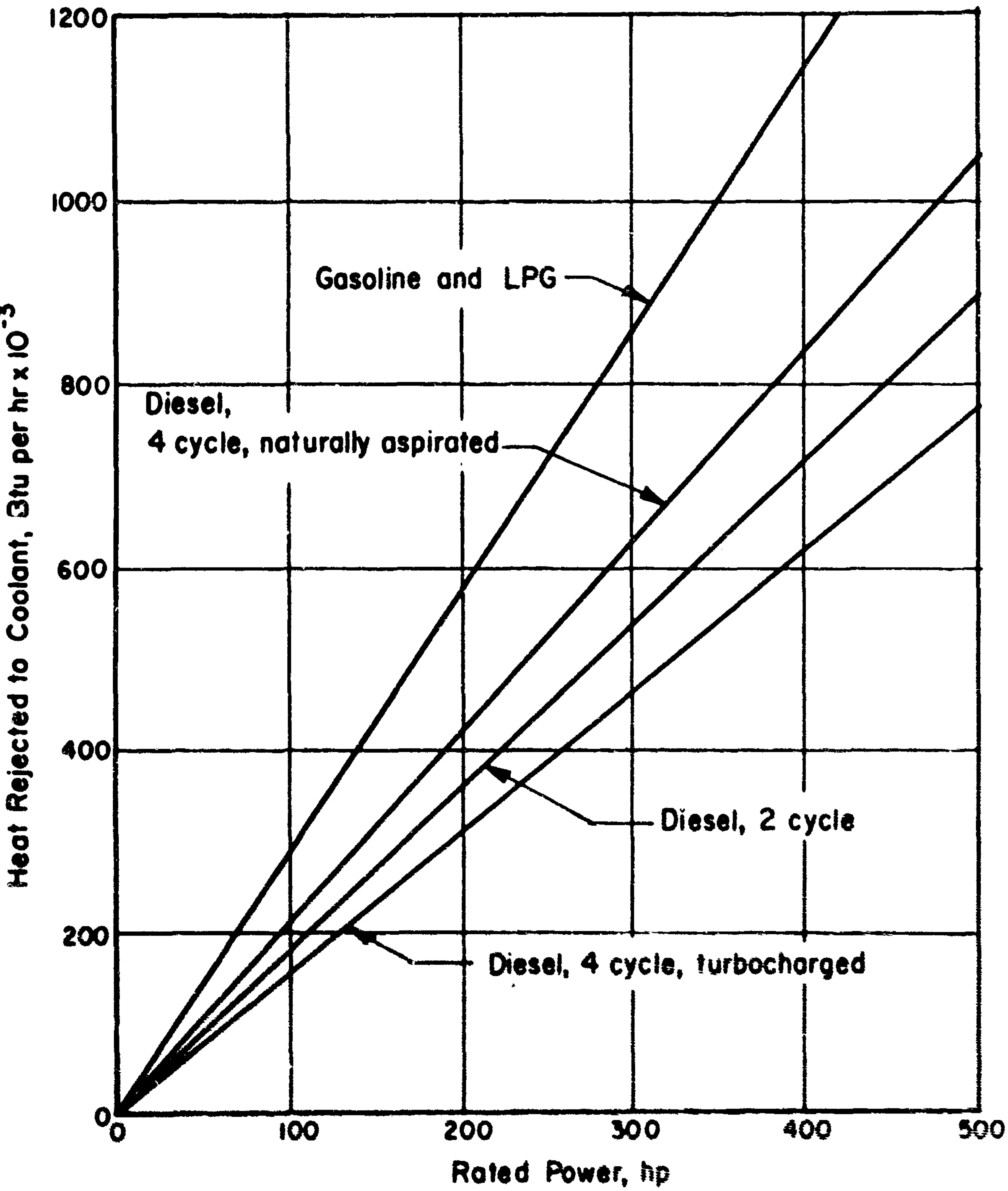


FIGURE 30. APPROXIMATE HEAT REJECTION INTO THE COOLING SYSTEM FOR VARIOUS PRIME MOVERS

Ventilation of Power System Enclosure

Even if air is not required to remove the waste heat rejected to the jacket water, a ventilation system for the engine room will be required. This ventilation system not only may be required to remove a significant quantity of heat from the enclosure, up to 50 per cent of the heat rejected to the primary cooling system, but it must also provide the air for combustion in the prime mover. A typical 100-hp auxiliary power system could require as much as 2,000 cfm of ventilating air for the power system enclosure. Approximately 25 per cent of this air would be necessary for combustion and the remainder would be needed to carry away heat rejected directly to the enclosure.

If the prime mover jacket water cooling system does not reject its heat to the engine room ventilating air, a separate blower must be provided. This blower should be designed with ducting, if necessary, to create an air circulation across the engine and any other heat-producing components in the enclosure. The exhaust or discharge opening from the power system enclosure should be so located that the cool incoming ventilation air is not short-circuited. If the jacket water cooling system utilizes any components within the enclosure which require cooling air, such as a radiator or a condenser, the ventilating air requirements for the power system enclosure will be significantly greater than 2,000 cfm.

It could be costly to provide total filtration for the power-system-enclosure ventilating air. Fortunately, the components of the auxiliary power system are not expected to be significantly affected by even large quantities of fallout radiation during the relatively short length of time covered by the emergency period. Furthermore, a properly designed auxiliary power system should require no maintenance or service during this operating period. Consequently, if the power system enclosure is effectively sealed from the occupied section of the shelter, its ventilating air will not have to be filtered for fallout.

Ventilating the power system enclosure with exhaust air from the occupied shelter space would be a very desirable method in that little, if any, additional equipment would be required. In the design of such a system, provisions would have to be made to insure that an adequate supply of air would always be available to the engine.

Cooling Systems

Disposal of the waste heat collected by the jacket coolant can be accomplished in a number of ways. These are:

(1) Direct make-up
(2) Heat exchanger
(3) Ebullient (boiling)
(4) Radiator.

A radiator system, being air cooled, would require no water except for initial filling of the system. Where the supply of water is limited, an ebullient system could be used. With an ample supply of water, the heat

exchanger or the direct make-up system could be selected. The direct make-up system is the simplest and least expensive. Following is a description of each of the systems.

Direct, Make-Up-Water Cooling

Figure 31 is a schematic illustration of a direct, make-up-water cooling system. In terms of the compounds required, this system is the simplest

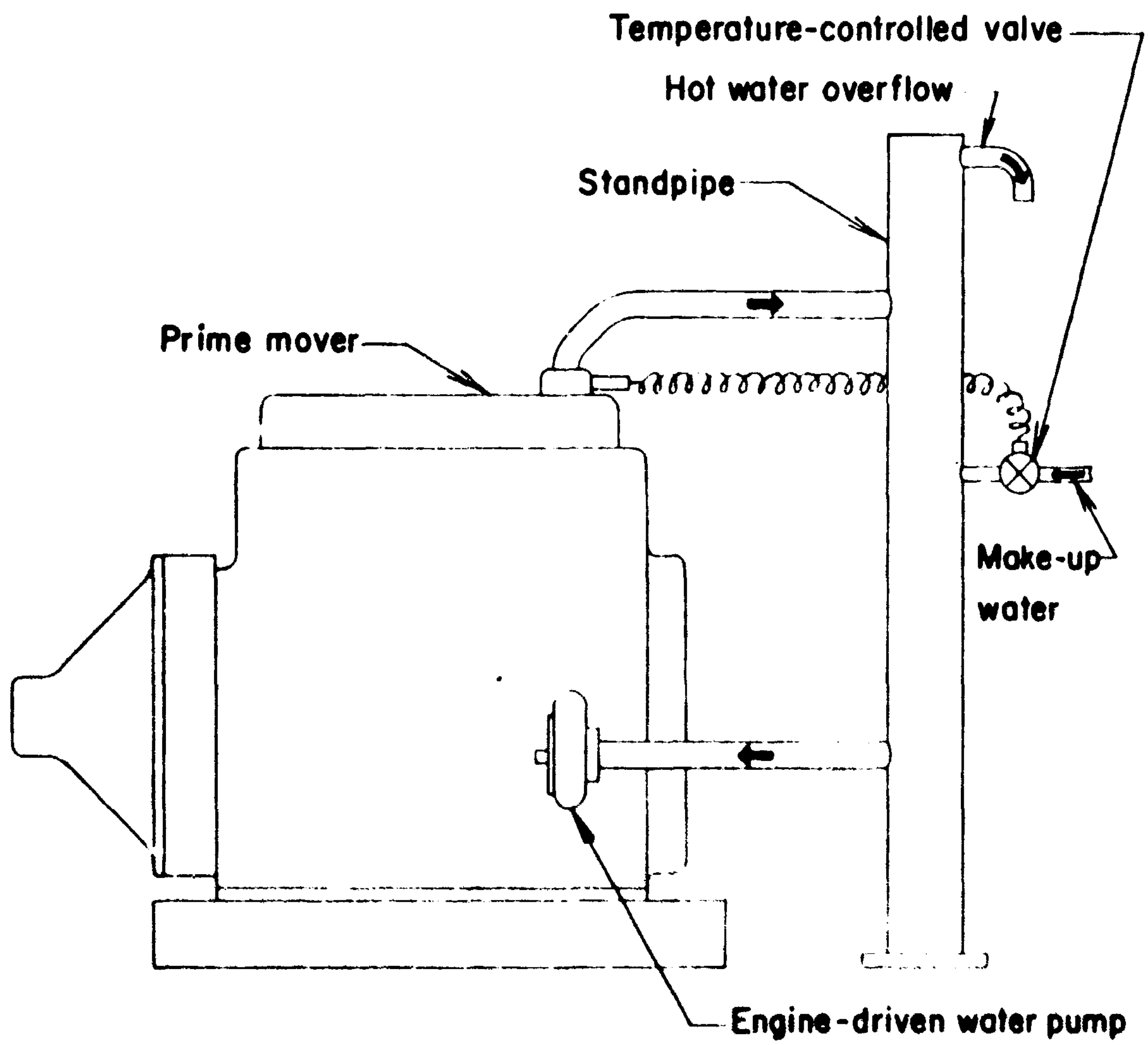


FIGURE 31. DIRECT MAKE-UP-WATER COOLING SYSTEM

and least expensive. As the illustration indicates, only two major components are required, a stand pipe and a temperature-controlled valve. The stand pipe could be a large-diameter pipe with a base plate welded at one end and with appropriate connections for hot water from the engine, cooled water to the engine, hot water overflow, and make-up water. The engine-driven water pump circulates the water through the system, and a temperature-sensing element in the hot-water discharge line from the engine controls the make-up water flow. Hot water is forced out the overflow in direct proportion to the amount of cold make-up water added.

The temperatures in an engine equipped with a system of this type would be nearly ideal. However, this type of system is very extravagant in terms of water usage. For instance, the water required for a 100-hp power source operating at full load is estimated to be about 6500 gal per day.

The installation of a direct, make-up-water cooling system would be extremely simple and the maintenance requirements would be almost negligible. For stand-by storage the system would be dry to prevent corrosion or sludge formation. The system would be extremely reliable and relatively insensitive to extreme ambient temperatures as long as a sufficient supply of make-up water was available.

A direct, make-up-water cooling system could be even further simplified by introducing the make-up water directly into the engine, thus eliminating the water pump and stand pipe. However, the adverse effect of introducing cooling water directly to the engine at 60 F and discharging it at 180 F on engine life and performance could be very serious. Most engines are not designed for such large temperature differentials and the thermal distortion could cause cracking of the engine block and engine failure. With part of the engine at low temperature where the cooling water first enters, there might also be high rates of corrosion and sludge formation in the lubrication system. Consequently, serious loss of power after only a relatively short period of operation could be expected.

Engine manufacturers in general do not recommend using raw make-up water directly in their engines. Most water, as it comes from the source, contains a considerable amount of dissolved and suspended solids (measured as "hardness") which tend to precipitate out and deposit on any surface with which the water comes into contact. The rate of deposition is accelerated by heating the water. Thus, if raw water is pumped through the cooling passages of an engine, in time all critical heat-transfer surfaces will be coated with deposits to such an extent that the cooling system will no longer function effectively. The rapidity of the coating process depends primarily on the hardness of the water and on the design of the engine cooling passages.

Chemically "softening" the water before using it in the engine will lengthen the effective operating life. However, when large quantities of water are involved, commercial softening equipment is quite bulky and expensive. It is possible that most engines could be operated for a two-week period on direct, make-up-water cooling without significant performance depreciation if the make-up water is not extremely hard.

Heat-Exchanger Cooling

Figure 32 is a schematic illustration of a heat-exchanger cooling system which includes a cooling tower to conserve water. The basic components of this sytem are: (1) heat exchanger, (2) surge tank, (3) temperature-controlled valve, (4) circulating pump, and (5) cooling tower. As indicated in the sketch, the cooling tower and circulating pump could be eliminated but at the expense of higher make-up water consumption. A vent line is provided between the engine discharge water line and the top of the surge tank to remove any noncondensible gases that may be trapped in the cooling water as it leaves the engine.

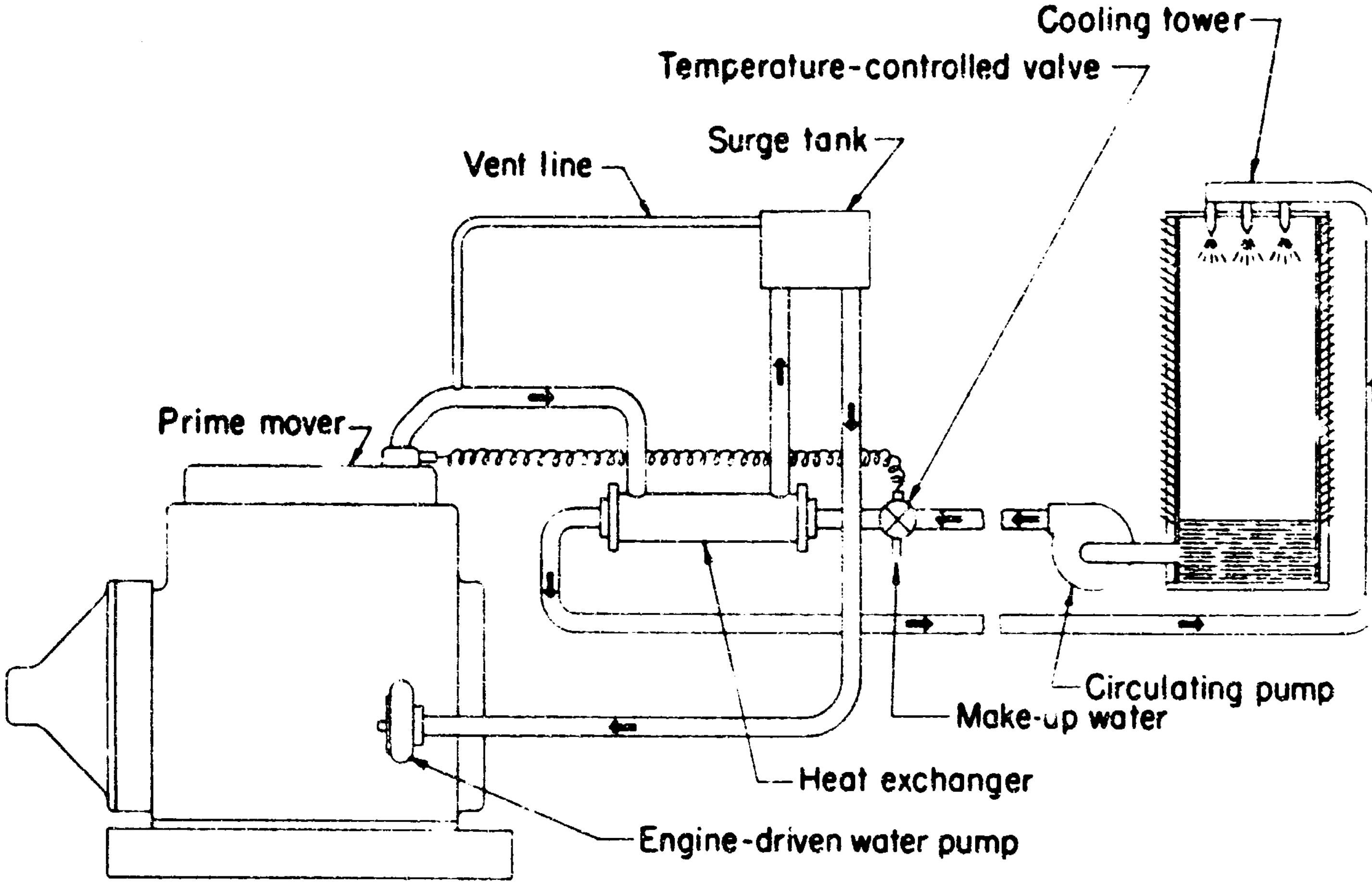


FIGURE 32. HEAT-EXCHANGER COOLING SYSTEM WITH COOLING TOWER

The make-up water requirement for a heat-exchanger cooling system without cooling tower is calculated to be about 4,750 gal per day for a 100-hp power source operating at full load. Use of a simple spray-type cooling tower would reduce the water requirement to about 850 gal per day, and a mechanical-draft-type cooling tower would reduce the water requirement to about 650 gal per day.

Certain shelter locations may lend themselves to the use of cooling ponds rather than cooling towers. The water consumption of a quiescent cooling pond of sufficiently large surface area for the heat rejection load of a 100-hp engine would be about 460 gal per day. Using a spray pond would reduce the water surface area required but would result in higher water consumption, approximately 1800 gal per day for the 100-hp engine.

Installation of the basic heat-exchanger cooling system would be relatively simple. The heat exchanger and surge tank should be located fairly close to the prime mover, and the piping should be so constructed that the water can be completely drained from the system when desired. The cooling tower would most logically be located outside of the shelter and be so positioned that the natural circulation of air through and around the tower would not be impeded. The piping system for the cooling tower should also be so constructed that the water can be completely drained from the system when desired.

The stand-by maintenance requirements of the heat-exchanger and cooling tower systems would be similar to those for direct, make-up-water cooling systems. If the heat-exchanger cooling system is not to be exercised regularly, it would be advisable to drain all the water out and seal all the components including the cooling tower after they are thoroughly dried out. If periodic exercising is planned, the heat-exchanger cooling system would require periodic inspection, replacement of the water supply, and cleaning of all critical components. Particular attention should be given to the cooling tower spray nozzles which cannot be allowed to clog up.

If installation and maintenance procedures are proper, the reliability of a heat-exchanger and cooling tower cooling system could be expected to be very good. The efficiency of a cooling tower is somewhat dependent on the ambient air temperature; consequently, the cooling tower would have to be sized for the highest temperature conditions anticipated.

Radiator Cooling

Figure 33 is a schematic illustration of two radiator cooling systems. This illustration shows the equipment and piping required for both an engine-mounted radiator cooling system and a remote radiator cooling system. The engine-mounted radiator cooling system is the familiar automotive-type system with an engine-driven water pump circulating the water through the engine and through a radiator, and with an engine-driven fan forcing air through the radiator. If a remote radiator must be located more than 15 ft above the engine level, a hot-well tank should be incorporated in the system between the remote radiator and the engine. The primary purpose of this hot-well tank is to reduce the pressure head on the engine since most gaskets and seals in the cooling system would not be designed to withstand high pressures. For this reason, the hot-well tank should be located just slightly above the level of the engine. In addition to a hot-well tank, a remotely located radiator cooling system would also require a second water pump to circulate the coolant through the radiator and an electrically driven fan mounted on the radiator. A remote radiator located less than 15 ft above the engine would probably not require a hot-well tank nor an extra circulating pump.

Engine temperatures in a radiator cooling system are controlled by a by-pass thermostat in the discharge water line from the engine. This by-pass thermostat causes water to be recirculated within the engine block until the desired operating temperature has been reached. At this time the radiator cooling system requires no external supply of water once the system is fully charged. However, a large amount of cooling air is required.

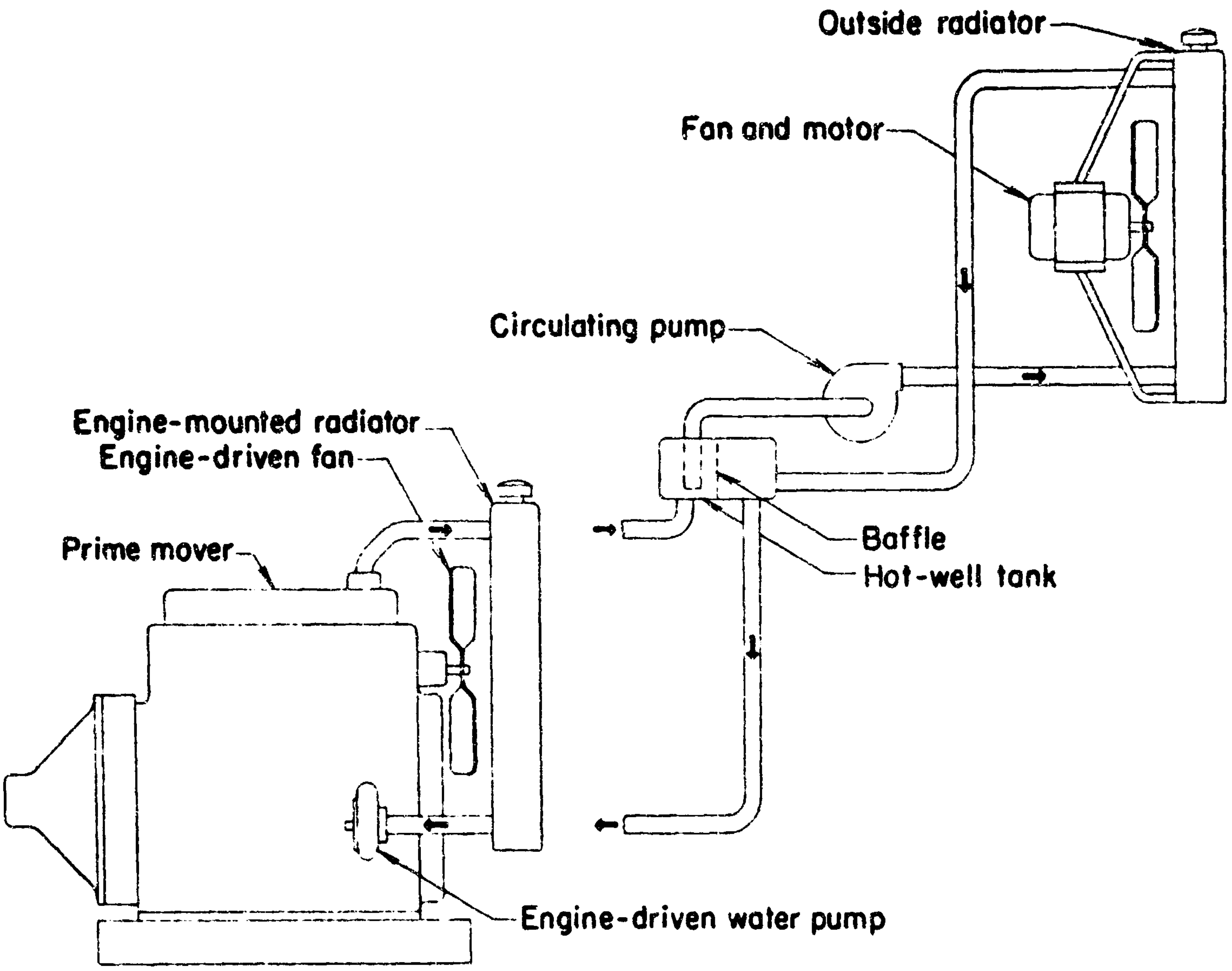


FIGURE 33. RADIATOR COOLING SYSTEMS

Figure 34 shows the approximate cooling air requirements of radiator-cooled prime movers. These data were obtained from manufacturers' literature and generally represent an excess cooling capacity. It is quite possible that the fan could deliver about 30 per cent less cooling air without engine overheating under normal operating conditions.

When a prime mover is installed with an attached radiator cooling system, the prime mover should be located with its radiator close to the enclosure exhaust opening. Moreover, a completely enclosed duct should be provided between the radiator and the exhaust opening to prevent recirculation of the hot air passing through the radiator. Assuming this kind of an installation and that the engine-driven fan is slightly oversized, the prime mover will be capable of providing its own cooling and combustion air flow providing that

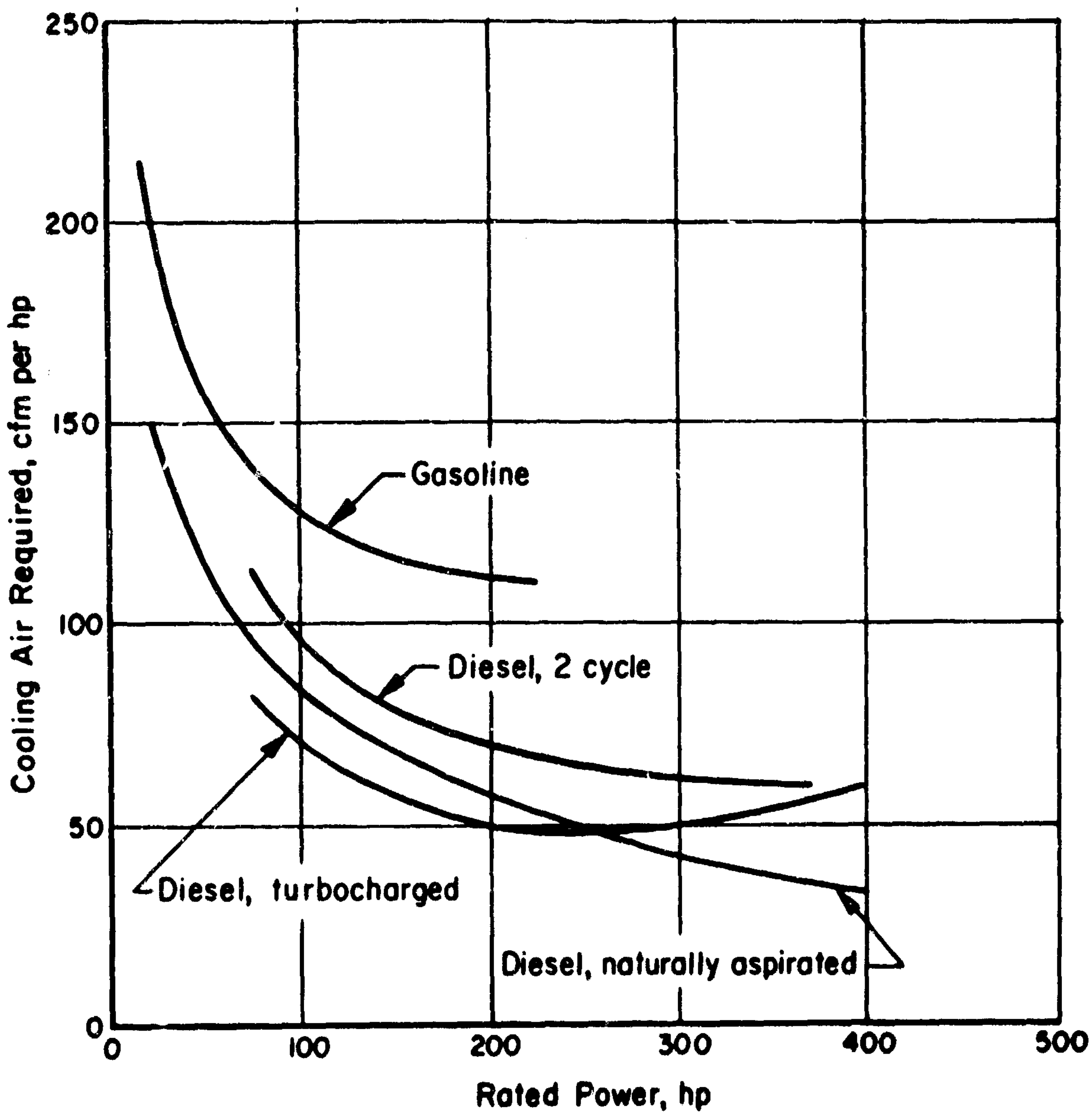


FIGURE 34. APPROXIMATE COOLING-AIR REQUIREMENTS FOR RADIATOR-COOLED PRIME MOVERS

restrictions in the inlet and exhaust systems for the auxiliary power system enclosure are minimum. If blast valves and/or total filtering systems are used, a separate blower will be required to assure proper ventilation in the enclosure.

Radiator cooling systems require little or no stand-by maintenance. If frequent exercising of the power system is planned, water should be allowed to remain in the system with suitable rust and corrosion inhibitors added to it.

The water should be sampled during each exercising or inspection and the system should be drained, flushed, and refilled when any indications of corrosion are observed.

The radiator cooling system would be a reliable system, but it would also be somewhat sensitive to extremes in ambient temperature. The effectiveness of radiator heat rejection is reduced as the temperature of the air passing over it increases. Where the ambient temperature is expected to go below freezing, the radiator should be equipped with a means to restrict the air flow; if the radiator is located outside the shelter, an anti-freeze solution should be added to the water.

Ebullient Cooling

Figure 35 is a schematic illustration of an ebullient or evaporative-type cooling system which utilizes the high latent heat of vaporization of water. This system consists basically of: (1) steam separator tank, (2) stand pipe, (3) steam vent, (4) water-level-control mechanism, and (5) level-control valve. The ebullient cooling system is based on the principle that increasing the pressure of water increases its boiling temperature. Water can be heated to a temperature of 220 F without boiling under a pressure head of about 6 ft of water. Consequently, if the water level in the steam separator is 6 ft above the top of the engine, the cooling water leaving the engine can be at a temperature of 220 F without boiling occurring. As this cooling water moves up the discharge line, the pressure is reduced and boiling begins. Because of the flow rate maintained in the system by the engine-driven water pump, not all of the water will vaporize before it reaches the separator. The steam in the separator is vented to the atmosphere and the water is recirculated through the engine. The coolant which is vented must be replaced by make-up water to maintain a constant water level.

An ebullient cooling system is estimated to require about 400 gal per day of make-up water for a 100-hp prime mover operated continuously at full load.

The installation of an ebullient cooling system would be relatively simple. The water level controls in an ebullient cooling system could be a potential source of trouble, as any corrosion tends to lead quite quickly to malfunction of this type of mechanism. For frequent exercising, the water level control should be inspected each time and cleaned if signs of corrosion are observed. For long-term storage this system should be dry with a thin coating of grease or oil on all critical parts.

With proper maintenance of the critical components, an ebullient cooling system should be very reliable. This type of cooling system is relatively insensitive to either high or low ambient air temperature. The ebullient cooling system provides nearly ideal operating temperatures and, therefore, would promote better engine performance and long engine life.(7)

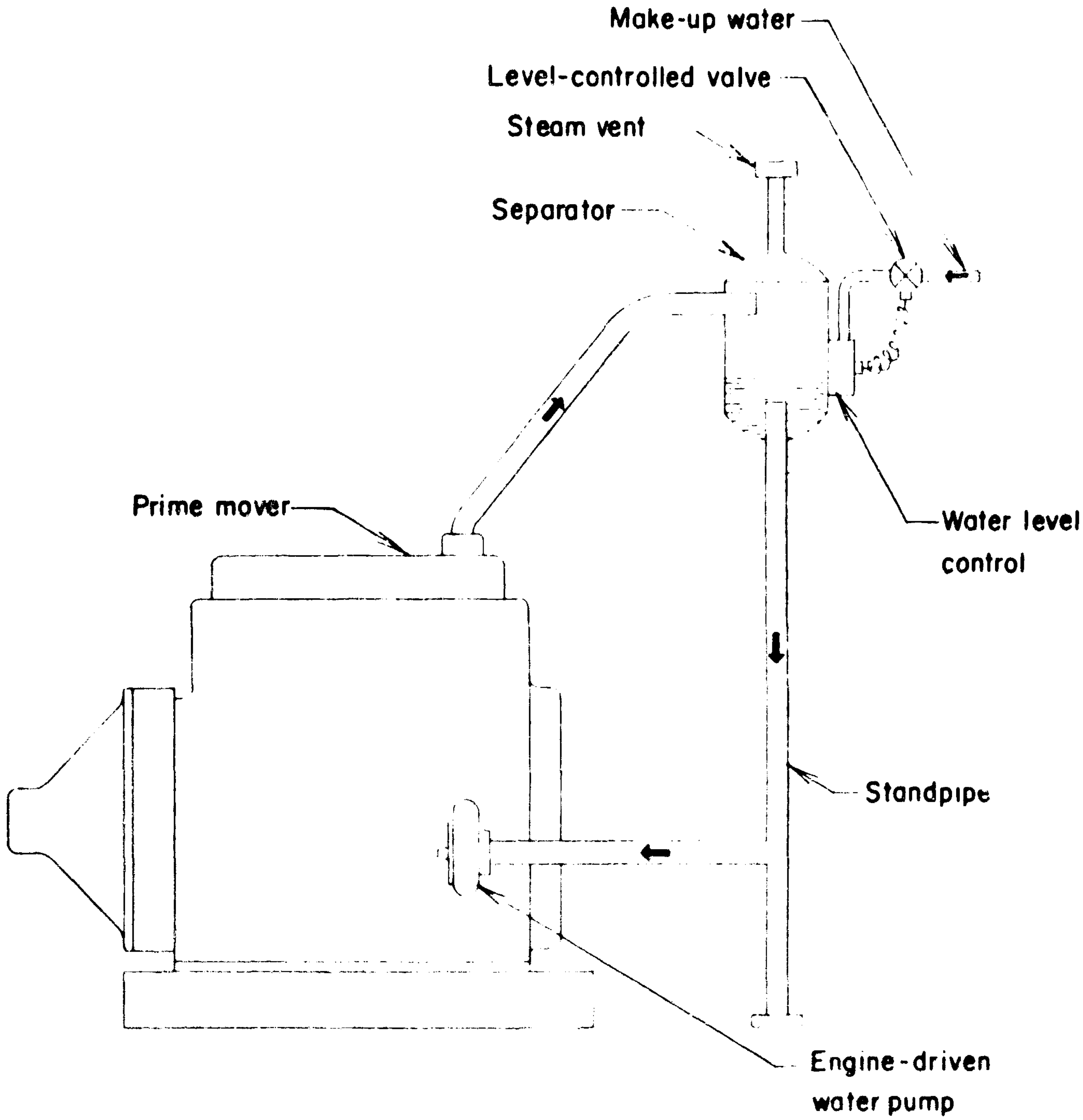


FIGURE 35. EBULLIENT COOLING SYSTEM

Comparison of Cooling Systems

Figure 36 shows estimated water requirements for the various cooling techniques discussed in this section. If an unlimited supply of cool water is available the direct make-up water system would be the best choice. It would be inexpensive, reliable, maintenance-free if stored dry, extremely compact, and require no cooling air. If the water is unsuitable for direct use in the engine, a heat exchanger which would add materially to the cost of the system could be added.

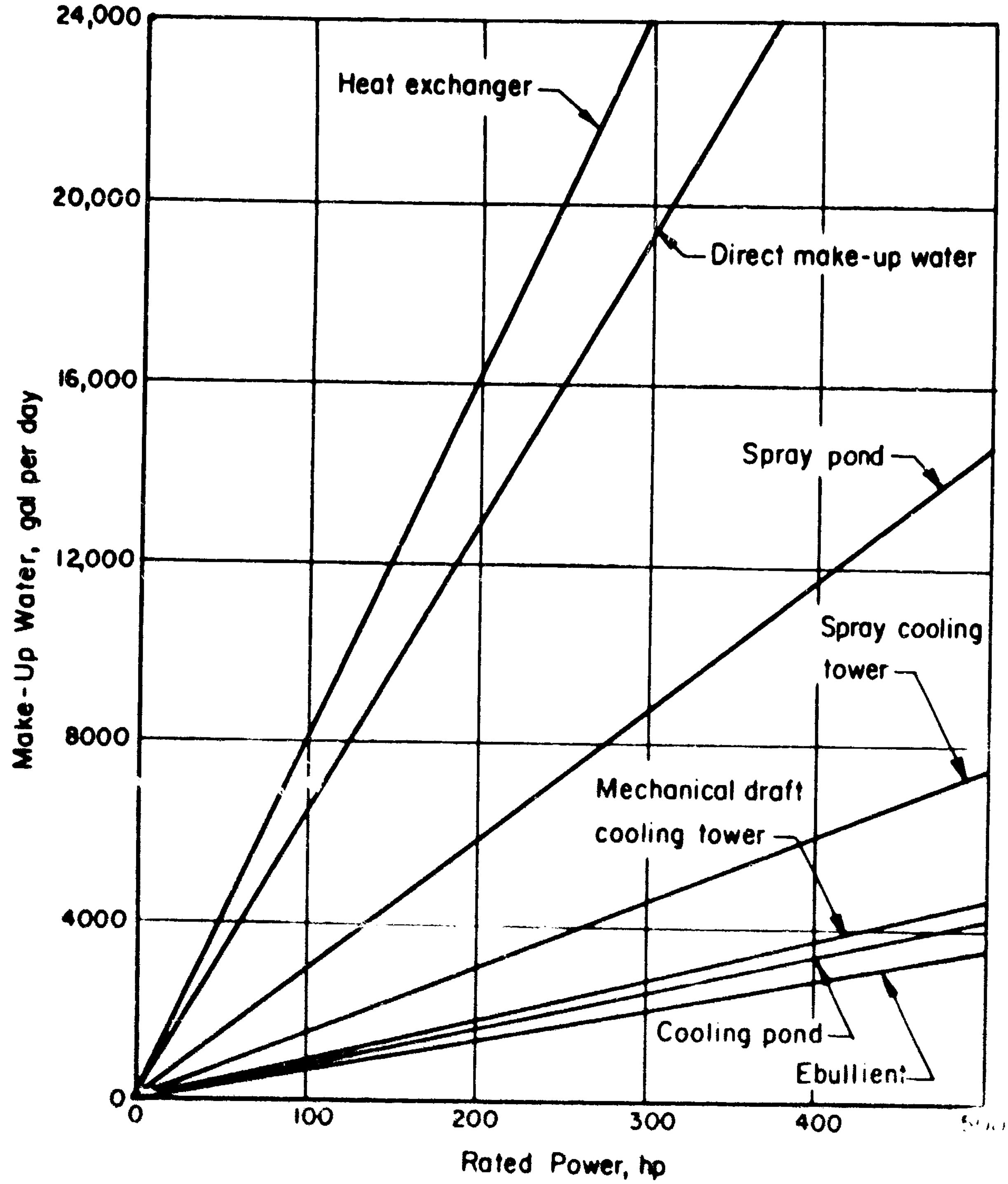


FIGURE 36. ESTIMATED MAKE-UP WATER REQUIREMENTS FOR VARIOUS COOLING TECHNIQUES

If a limited amount of water is available, an ebullient system can be used. This system also requires more equipment than the direct make-up-water system and, therefore, is more costly. However, it requires the least amount of make-up water.

An advantage of the above systems is that they are completely contained in the shelter and, therefore, are not subject to damage by blast, fire, etc.

The cost of a radiator cooling system using air as the heat sink would be about the same as the cost of a heat-exchanger water-heat-sink cooling system. The engine-mounted radiator system would be lower in cost than the externally mounted radiator especially if the external radiator had to withstand the effects of blast, fire, etc.

FUEL STORAGE

Fuel stored for use by the prime mover should be adequate, both in quality and quantity, for at least two weeks of continuous full-load operation. In addition, it may be necessary that fuel be supplied to the prime mover during periodic exercising. Usually engine fuels are not stored for more than a few months and, most commercial engine fuels deteriorate during extremely long storage.

This discussion will cover four common petroleum fuels: 90 octane gasoline, kerosene, No. 2 diesel fuel, and liquified petroleum gas (LPG). These fuels can be considered as representative of the wide variety of petroleum fuels available. The designer has available at least two of these fuels, regardless of the power source selected. As previously mentioned, the prime movers considered most practical for shelter use are compression-ignition engines, spark-ignition engines, and gas turbines. Compression-ignition engines may burn either kerosene or No. 2 diesel fuel; spark-ignition engines may burn either gasoline or LPG; and gas turbines may burn any of the fuels listed including leaded gasoline if special precautions are taken.

As fuels deteriorate during storage, they eventually reach the point where they are no longer useable in engines. The length of time before a fuel becomes unuseable depends on several factors: (1) the quality of the fuel when fresh, (2) the storage technique, (3) the characteristics of the engine in which it is intended that the fuel be used, and (4) the length of time the engine must operate. To ensure a supply of useable fuel, the designer has the choice of two basically different approaches: long-term fuel storage with special precautions being taken, or frequent replacement of a conventionally stored fuel.

In the design of fuel storage systems for community shelters, it is necessary to consider the possible influences of: natural weather conditions, blast pressure or ground shock, initial gamma radiation, neutron flux, and thermal radiation resulting from nuclear explosion. In addition, everyday factors affecting storability are: chemical and physical changes in the fuel caused by evaporation, oxidation, and polymerization; contamination of the fuel by condensed water vapor, air-borne dirt, bacteria, and materials resulting from storage tank deterioration; natural weather and soil conditions which may cause mechanical failure of the storage tank or connecting fuel lines.

Prime Mover Fuel Requirements

Engine suppliers usually specify one or more commercially available fuels for use in their engines. These fuels, when fresh, will have a tolerable gum content, an acceptable octane or cetane rating, and a vapor pressure adjusted for warm or cold weather use. The objective of long-term storage of fuels is to ensure that the desired fuel properties are preserved. Each type of engine has its own fuel requirements which are discussed in detail in the following paragraphs.

Compression-Ignition Engines

Fuel for a compression-ignition engine should meet standards of: cetane rating, vaporization characteristics, sulfur content, corrosion properties, gum content, and carbon residue. Cetane rating is an indication of the ignition quality of the fuel. An adequate cetane rating is necessary to enable the engine to deliver full power. The cetane ratings of commercial diesel fuels vary widely. Ratings between 40 and 60 are acceptable for most diesel engines. Fuels with values below 40 cause engine smoking and loss of power. For a particular engine, the manufacturer should be consulted regarding exact fuel requirements.

Loss of the lighter fractions of the fuel, a potential result of long-term storage, may seriously affect vaporization characteristics. The principal result will be that the engine will exhibit poor cold-starting characteristics. However, there may also be some depression of the cetane rating of the fuel. If the fuel remains adequate for full-load engine operation, any decrease in cold-starting ability because of depreciation of vaporization characteristics can be compensated for by the use of a cold-starting aid such as ether sprayed into the air intake system. Several compounds are commercially available for this purpose.

ASTM Standard D975-T for diesel fuel allows a maximum sulfur content of 1.0 per cent by weight. Sulfur contents of 0.5 per cent or less are preferred for an engine which is idle for long periods of time. A sulfur content of 0.5 per cent is common for premium diesel fuels. The ASTM Standard for kerosene is based on its burning qualities in an illuminating lamp. No maximum sulfur content is given; however, the burning-quality requirement indirectly places limits on the sulfur content. As a result, sulfur contents of less than 0.1 per cent are common in kerosene. The corrosiveness of the fuel is related more to the chemical form of the sulfur than to the total sulfur content. A corrosive fuel is unacceptable because of the possibility of damage to the close fitting parts of the engine, the fuel pump, and the injectors as well as to the storage tank and fuel supply lines.

Sulfur dioxide or sulfur trioxide formed in the engine combustion chamber will be largely expelled with the engine exhaust. Oxides of sulfur will combine with water vapor and condense to form sulfurous acid if the temperature of the exhaust gases falls below approximately 300 F. Accordingly, sulfur could cause corrosion problems in the exhaust system if the engine is operated for too short a time during exercising, or the exhaust gases are cooled below 300 F by a waste-heat recovery system in the exhaust. It is also possible that oxides of sulfur could contaminate the lubricating oil and contribute to varnish formation on the cylinder walls.

ASTM Standard D975-60T for diesel fuel does not list a maximum gum concentration level. The soluble gum does not appear to be directly harmful to operation. The liquid fuel, along with the dissolved gum, is injected directly into the combustion chamber. The gum burns, along with the fuel, and the products of combustion are exhausted from the system.

Insoluble gum, which does not settle out in the main fuel tank, is filtered out, to a large degree, before it reaches the critical parts of the fuel system. The insoluble gum which does pass through the filter system is in the form of finely divided particles, 5 microns or less in diameter. These gum particles will contribute to galling of the precision parts in the system. It will also contribute to carbon formation in the combustion chamber. However, the small amount of gum which will pass a good filter should be tolerable.

Carbon residue is that part of the fuel that remains when the heavy fractions, including gum, are burned with a limited supply of oxygen. Further oxidation of the carbon residue leaves an ash. Under full-load operating conditions, the detrimental effects of carbon residue are minimized. Under all operating conditions, the effect of ash on engine wear is slight.

Spark-Ignition Engines

Fuel for a spark-ignition engine should meet standards for octane rating, gum content, sulfur content, corrosion properties, and vaporization characteristics. The octane rating of a spark-ignition engine fuel is a measure of the anti-knock characteristics of the fuel. As a rule, the octane requirement of an engine increases with load. Therefore, the fuel in storage must retain an octane rating suitable for full-load operation.

Operation of the spark-ignition engine requires that the fuel be vaporized before entering the combustion chamber. As a result, any nonvolatile constituents, such as dissolved gums, are largely dropped out in the carburetor, along the intake manifold, and on the intake valves and ports.

The petroleum fuels considered in this discussion which may be used in spark-ignition engines are gasoline and liquified petroleum gas (LPG). Spark-ignition engines designed specifically to operate on LPG are available. Also, gasoline engines may be modified, with commercially available kits, for LPG operation. However, these kits do not change the octane requirements of the engine. Consequently, the gasoline engine modified for LPG operation cannot derive maximum benefits from the higher octane qualities of the LPG fuel.

ASTM Standard D439 for gasoline specifies a maximum gum content, soluble and insoluble, of 5 grams per 100 liters. This is considered tolerable in spark-ignition engines. Gum particles will be deposited when the gasoline is evaporated. However, a light oil, which is usually mixed with the gasoline, will also be deposited and will act to wash the gums along the distribution system and into the combustion chamber. As a result, gum from the fuel is ultimately burned and exhausted from the system. The standard of 5 grams of gum per 100 liters of gasoline should be a safe level. Work done by Southwest Research Institute[8] indicates that gum contents as high as 7 grams per 100 liters may be tolerated for 350 hours of operation. However, this cannot be considered as an absolute value for all spark-ignition engines, because the tolerable gum content is a function of the nature of the gum, engine design, the quantity of light oil mixed into the gasoline, and the engine exercise program.

A critical factor affecting the reliability of gasoline-engine-powered emergency stand-by equipment is the formation of gum in the carburetor jets during idle time. When an engine is not operating, slow evaporation of the fuel from the carburetor jets and the float chamber results in gum deposits at the point of evaporation. This is one reason that weekly exercising of gasoline engines is frequently recommended. For longer periods of non-use, the fuel should be removed from the carburetor.

Sulfur content as high as 0.25 per cent is considered acceptable in most federal gasoline specifications. Because an emergency stand-by power plant is idle most of the time, it is more susceptible to corrosion attack than regularly operated

engines. For this reason, a sulfur content of 0.1 per cent or less is desirable. Sulfur dioxide or sulfur trioxide formed in the combustion chamber will be largely expelled with the engine exhaust. Oxides of sulfur will condense to form sulfurous acid if the temperature of the exhaust gases drops below 300 F. Therefore, sulfur could cause corrosion problems in the exhaust system if either the exercise period is too short to fully warm up the engine or the exhaust gases are cooled below this temperature in a waste-heat recovery system. It is also possible that oxides of sulfur could contaminate the lubricating oil and could contribute to varnish formation on cylinder walls.

As in the case of the compression-ignition fuels, long-term storage of gasoline will result in loss of the lighter fractions which can seriously affect the vaporizing characteristics causing poor cold starting. As long as the gasoline is adequate for full-load operation, however, any cold-starting difficulties can be compensated for by the use of highly volatile cold-starting aids which are commercially available for gasoline engines as well as for diesel engines.

ASTM Standard D1835-61T for liquified petroleum gas (LPG) calls for a maximum sulfur content of 15 grams per 15 cubic feet of gas. Propene (propylene), an olefin, is a common contaminant in LPG. Although propene is stable in storage, its octane rating is lower than either propane or butane, the primary constituents of LPG, and its burning characteristics are inferior. For these reasons, the propene in LPG should be limited to 5 per cent. Spark-ignition engines operating on propane-butane mixtures should exhibit clean-burning as well as gum-free operation.

Gas Turbines

The fuel requirements for gas turbines are much less stringent than those for compression-ignition and spark-ignition engines. The gas turbine is capable of burning any of the commercially available petroleum fuels. However, the use of leaded gasoline is not recommended. Leaded gasoline will cause a high rate of lead oxide deposition on the turbine blades, in the combustion chamber, and in the regenerator. This seriously shortens turbine useful life and could also reduce power output and efficiency.

Fuel Deterioration

As a rule, evaporation, oxidation, polymerization, and contamination all have a detrimental effect on the quality of most petroleum fuels during storage. These result in the formation of gum, gel, and sludge. The reactions are interrelated, complex, and difficult to predict.

Figure 37 shows typical atmospheric-pressure distillation curves for the liquid petroleum fuels considered in this discussion.[9] Because gasoline, kerosene, and No. 2 diesel fuel are made of a variety of hydrocarbon components, evaporation of the lighter fractions of the fuel over a period of time will result in an increased concentration of the higher-boiling-point components. Consequently, gradual evaporation during storage modifies the composition of the fuel. The evaporation problem is most severe for gasoline.

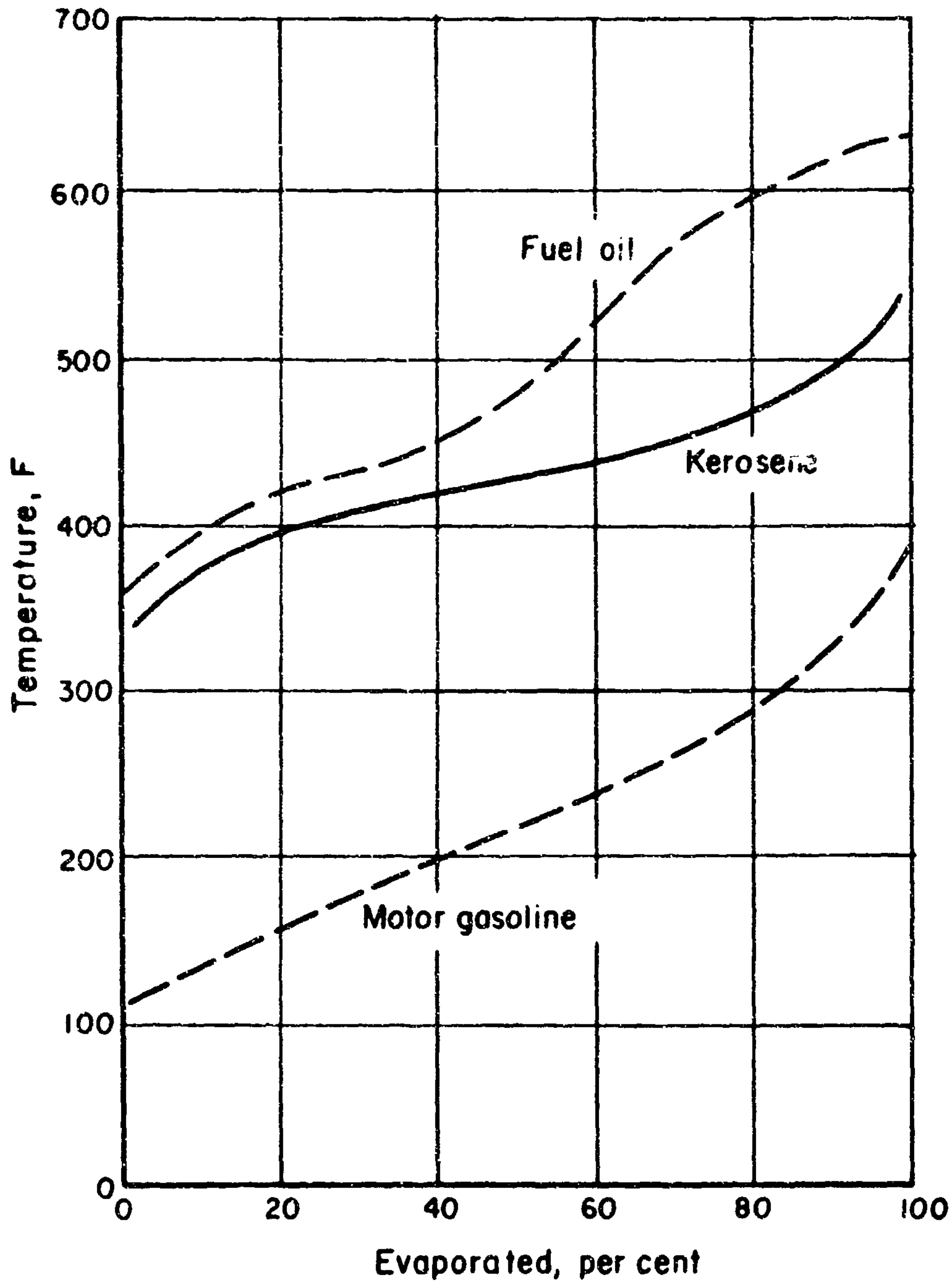


FIGURE 37. TYPICAL ASTM DISTILLATION CURVES FOR PETROLEUM FUELS

Some degree of evaporation in a vented tank is to be expected because of "breathing" and diffusion. Breathing is caused by the expansion and contraction of the tank contents due to temperature changes. Also, expansion and contraction of the vapor-air mixture at the top of a vented tank due to changes in barometric pressure and wind velocity is a factor. Above-ground storage tanks are subjected to more frequent and much greater temperature changes than underground tanks. The temperature of the fuel in an above-ground tank may vary as much as 100 F throughout the year, whereas in an underground tank total temperature variations of about 40 F are more typical. The fuel storage system should be designed either to eliminate or greatly retard evaporation.

Of the many chemical reactions which may take place in stored fuel, partial oxidation and polymerization are most important. Of these two, oxidation is the predominant reaction. Unsaturated molecules, primarily olefins, and impurities in the fuel will, in time, polymerize or react to form various products, some of which could adversely affect engine operation. These reaction products are commonly called gum, sluoge, and gel.

The gummy substance formed as a result of oxidation and polymerization is called, appropriately enough, gum. Gums may be either soluble, and thus not detectable until the fuel is evaporated, or insoluble, and thus removable by filtration. Slugde is a general term which is applied to all insoluble material which settles to the bottom of a fuel tank. Insoluble gums which stay suspended in the fuel are not considered part of the sludge. Insoluble gums which adhere to steel tank walls or other iron-alloy system components in contact with the liquid fuel are referred to as adherent gums.

No two batches of fuel coming from a given refinery are exactly alike in every respect. Likewise, fuels from different refineries are not alike. Experience has shown that in some instances different batches of fuel are not compatible. That is, when the two are mixed, the gum and sludge formation is greater than that which might be expected considering the original gum content of the two separate fuels.

Generally speaking, either straight-run fuels or completely hydrogenated cracked fuels, both of which contain a minimum of olefins, will offer the least potential for gum and sludge formation. Sludge formation is accelerated by high temperatures, oxygen, and the catalytic action of corrosion products. Water in a fuel storage tank will increase the amount of corrosion products which might contaminate the fuel.

The jelly-like deposit which forms on copper-base alloys and zinc is referred to as gel. Chemically, gels are largely copper and zinc mercaptides. Gels form as a result of a reaction between the mercaptans in the fuel and copper and/or zinc. Fuels low in mercaptans, less than 1 gram/liter of sulfur, in general, have low gel-forming rates.

Sulfur in a fuel, if in a certain chemical form, will cause harm by contributing to the corrosiveness of the fuel, contaminating the engine lubricating oil, or corroding the engine exhaust system. Fuel corrosiveness is a definite consideration in the fuel storage problem. A corrosive fuel will, if stored for a long period of time, not only significantly corrode the fuel system, but also contribute to gum and sludge formation through the formation of catalytic corrosion products.

It is possible, after long storage, for noncorrosive fuel to become acetic due to a reaction between the sulfur compounds in the fuel and oxygen. Should corrosion become evident, a corrosion inhibitor should be added and the fuel retested. Compatible corrosion inhibitors can be obtained from fuel suppliers.

The corrosiveness of a fuel can be determined through the ASTM D130-56 copper-strip corrosion test. Briefly, this test consists of immersing a freshly polished copper strip in a fuel sample and sealing the container. The sample is then heated after which the copper strip is removed and compared with a set of ASTM standard copper corrosion plaques.

Deposition of dirt in water condensed from vapor in the air occurs in vented fuel storage tanks because of the breathing of the tank. The water, which is heavier than the fuel, collects on the bottom of the tank and can be pumped out periodically. The presence of water in a tank may be detected by applying a substance commonly called "water indicating paste" to a dip stick and inserting the stick in the tank. Water is indicated by a change in the color of the paste.

Some fuels support bacterial growth. This is most pronounced in open tanks and in the presence of water. Additives suitable for retarding bacterial growth are available from the refiners and should be used, particularly if the fuel is to be stored in a vented tank for a long period of time.

The fuel can also be contaminated as the fuel-storage tank rusts from the outside in. For above-ground storage, periodic painting of the outside of the tank should be adequate. Proper installation of an underground storage tank will depend on local soil and water-table conditions. Conventional corrosion protection methods should be used for the storage tank.

Effects of Nuclear Weapons

The design of a shelter system must include protection of stored fuel from the primary effects of a nuclear explosion. These effects include one or more of the following: blast, thermal radiation, initial ionizing radiation, and ionizing radiation from fallout.

Blast

Air-burst nuclear weapons produce two pressure effects: overpressure and dynamic pressure. Overpressure is the pressure in excess of atmospheric which exists at the blast wave front. Dynamic pressure is the pressure resulting from the strong winds accompanying passage of the blast wave.

Above ground, the storage tanks must resist both pressure effects. This means that they have to be well anchored to resist dynamic pressure and ruggedly designed to resist crushing resulting f[illegible] pressure. Also, protection has to be provided to prevent damage by flying deb[illegible].

Under ground, the storage tanks would be affected primarily by overpressure. An exception is the tank vent which would also be affected by dynamic pressure.

Both the aboveground and underground storage tanks must be designed to prevent the vents from being damaged by bending, etc., to prevent overpressure from bursting the tank, and to ensure that the blast does not crack or rupture the tank.

Fuel tanks should be designed to withstand a higher blast pressure than the shelter proper, because cracked or torn seams in the shelter may not render the shelter unusable whereas an opening in the bottom or side of the fuel tank would cause power system failure. As the blast and shock waves pass some relative movement between the shelter and the fuel tank will occur due to downward displacement

and vibration. Adequate flexibility should be provided in the piping connecting the fuel tank to the shelter to allow for the relative movement.

Thermal Radiation

The technique for providing protection against thermal radiation is essentially one of providing enough thermal insulation to shield components of the fuel storage facility from the heat associated with the design blast pressure. At blast pressures of 100 psi, there will be intense heating of exposed vents, etc., due to thermal radiation and hot gases. Thermal radiation levels of about 600 calories per square centimeter and gas temperatures up to 1800 F are typically associated with this blast overpressure. The high heat flux period passes within 15 to 20 seconds[10] for a 1-megaton explosion. Consequently, although surface temperatures become quite high, the heat does not have time to penetrate very far before the wind associated with the blast wave begins to cool the heated surfaces. Thus, aboveground structures which are heavy and rugged enough to withstand the blast may be expected to tolerate the heat. Under most conditions, it should be possible to provide sufficient shielding to protect fuel from any noticeable temperature rise. However, in some situations, it may be necessary to plan for some temperature rise due to heat from either the direct effects of the nuclear explosion or the secondary fires resulting from the explosion. Fuels may be heated to 150 F for short periods of time (1 hour or less) without causing chemical breakdown or excessive gum formation. However, at this temperature, the lighter fractions in gasoline fuel would be lost due to "boil-off".

Initial and Fallout Radiation

Ionizing radiation resulting from a nuclear explosion can damage petroleum products if the radiation levels are sufficiently high. However, the radiation levles, both initial and fallout, associated with blast overpressures of 100 psi or less do not appear to be high enough to be of concern. Above 100 psi blast pressures, shielding against the effects of initial radiation, gamma rays, and neutron flux should be considered.

When petroleum fuels are exposed to ionizing radiation, chemical bonds are broken with the subsequent release of hydrogen and the possible polymerization to gum-forming substances or to substances which cause changes in viscosity. The following was taken from a report by the Esso Research and Engineering Company:[11]

"The studies show that unsaturated compounds are more reactive than saturates, and aromatic compounds are the most stable of all.....irradiation experiments with multicomponent systems indicate that data on individual organic compounds cannot predict the radiolysis behavior of complex mixtures of the type found in petroleum.

"A review of the available data on the radiolytic behavior of petroleum fractions reveals that they can usually be exposed to radiation dosages in the range of 10^6 to 10^8 rads without serious damage to properties affecting their end use. With jet fuels, for example, doses in the neighborhood of 10^7 rads have been observed to produce threshold damage effects. Petroleum-based lubricants can generally withstand several times this amo nt of radiation. Literature data on

diesel fuel, gasoline, and other petroleum fractions are meager; but it appears that they should be as stable as jet fuel unless they contain radiation-sensitive additives such as tetraethyl lead. Crude oil apparently is more radiation resistant than any of the products prepared from it.

"The available data on the evolution of gas from irradiation hydrocarbons show that the range of gas yields is 1 to 5 milliliters of gas per milliliter of liquid per 10^8 rads dosage. A consideration of the possibility that gamma rays and fast neutrons may differ in their effectiveness for causing radiation damage has led to the conclusion that in most cases serious errors will not be introduced by assuming equal damage from equal energy input."

A later report(12) confirmed the correctness of the above information on radiation damage.

Therefore, from the point of view of the petroleum products, it is recommended that the fuel storage tank and the power system enclosure be shielded to limit the exposure to not more than 1 million rads or approximately 1 million roentgens. This measurement of total exposure for a two-week period should include both gamma radiation and neutron flux. Initial radiation levels of 1 million rads will not be associated with blast pressures of 100 psi or lower. At a given blast overpressure, the associated initial radiation intensity increases as the weapon size decreases. Thus, after a design blast pressure has been selected, the shielding should be selected on the basis of the radiation characteristics of the smallest weapon likely to be used.

The gas released from 2,500 gallons of fuel (approximate two-week supply for 100-hp prime mover) when exposed to 10^6 roentgens would be about 20 cu ft, assuming the maximum release rate. This quantity would be equal to about 6 per cent of the total stored volume of fuel. Petroleum-base hydraulic fuels may be expected to react to radiation exposure in the same manner. Therefore, it is desirable to provide some excess storage capacity in all fuel and oil tanks so that the tank pressures will not exceed an allowable maximum.

Fuel Storage Techniques

Many techniques are available to the shelter designer for storing liquid fuels. Those techniques considered most applicable to community shelters will be discussed here. In selecting a storage technique, the important considerations are: prevention of deterioration, preservation of combustion qualities, and meeting the requirements imposed by state and local fire and safety regulations. Regarding the latter, it is reasonable to expect that in some instances it will be necessary to obtain special permission in order to use the most satisfactory storage system.

Commercial fuels are not formulated for long-term storage. There are two basically different fuel storage techniques which may be followed: active storage and long-term storage. In an active fuel-storage program a fuel would be replaced or replenished at regular intervals, and the storage tank requirements would be relatively uncritical. In a long-term fuel-storage program the storage system should be designed to preserve the fuel for the longest possible period of time, and the fuel quality should be checked at regular intervals so that the first signs of deterioration can be detected before there is serious degradation

of fuel qualities. The time intervals between inspection and/or replacement of the fuel would vary considerably with the type and quality of the fuel as well as with the storage conditions.

Table 6 shows approximate storage life for different fuels under different storage conditions. The values shown were arrived at by analysis of storage life data and opinions from a large number of organizations concerned with fuel storage. Unfortunately, there has been relatively little research done on the problem. As a result storage practices vary widely and tend to be conservative relative to opinions of maximum storage life. In view of the uncertainties regarding storage life and of the many variables involved the estimates given in Table 6 may be in error as much as 50 per cent for any specific situation.

Active Fuel Storage

Perhaps the simplest arrangement for ensuring an adequate fuel supply would be to connect into the fuel tank of a neighboring service station. If the amount of fuel which the service station would withdraw from the storage tank were limited, the tank would always contain at least the minimum quantity of fuel required for operation of the auxiliary power system. With this arrangement, some of the fuel in the tank would be continually withdrawn and replaced. As a result of frequent mixing and dilution by the fresh supply, the over-all quality of the fuel should remain adequate. Once a routine was established and checked, this storage arrangement should not require a detailed fuel monitoring program.

Another possibility along these lines is to install an oversized tank at the shelter and then to arrange for a private organization to use the fuel facility. A positive means should be provided to limit the amount of fuel that could be withdrawn by the user so that the community shelter minimum fuel supply would not be jeopardized by human error or otherwise. Again, the frequent withdrawal and replacement of fuel should ensure a supply of reasonably fresh fuel.

Figure 38 is a schematic illustration of an underground vented fuel-storage tank. The components are arranged so as to be consistent with typical code requirements. The vent should not be located next to a building or under the overhang of a building. Fire regulations may require that the fuel line between the tank and the shelter slope toward the fuel tank. If this is the case, the arrangement shown would have to be modified or special permission would have to be obtained. Permitting gravity feed of the fuel from the tank to the shelter does involve the risk of draining the entire fuel supply into the prime mover enclosure if the shutoff valve is accidentally left open and there is a leak in the piping system within the shelter. However, the simplicity of gravity feed is a significant advantage in terms of both cost and reliability, and close control over operation of the fuel system should minimize the risk of flooding.

Flexible sections are included in the fuel lines, both between the fuel tank and the shelter and between the shelter floor and the prime mover to reduce the danger of failure in the lines due to the relative movement of these components. A charging pump is provided in the system in the event that minor blockage in the fuel system should prevent normal gravity flow to the "day tank". The "day tank", mounted on the engine, reduces the number of fuel lines which must be run between the main fuel tank and the power source; provides for additional settling and straining of insoluble contaminants from the fuel; and provides the proper fuel pressure head

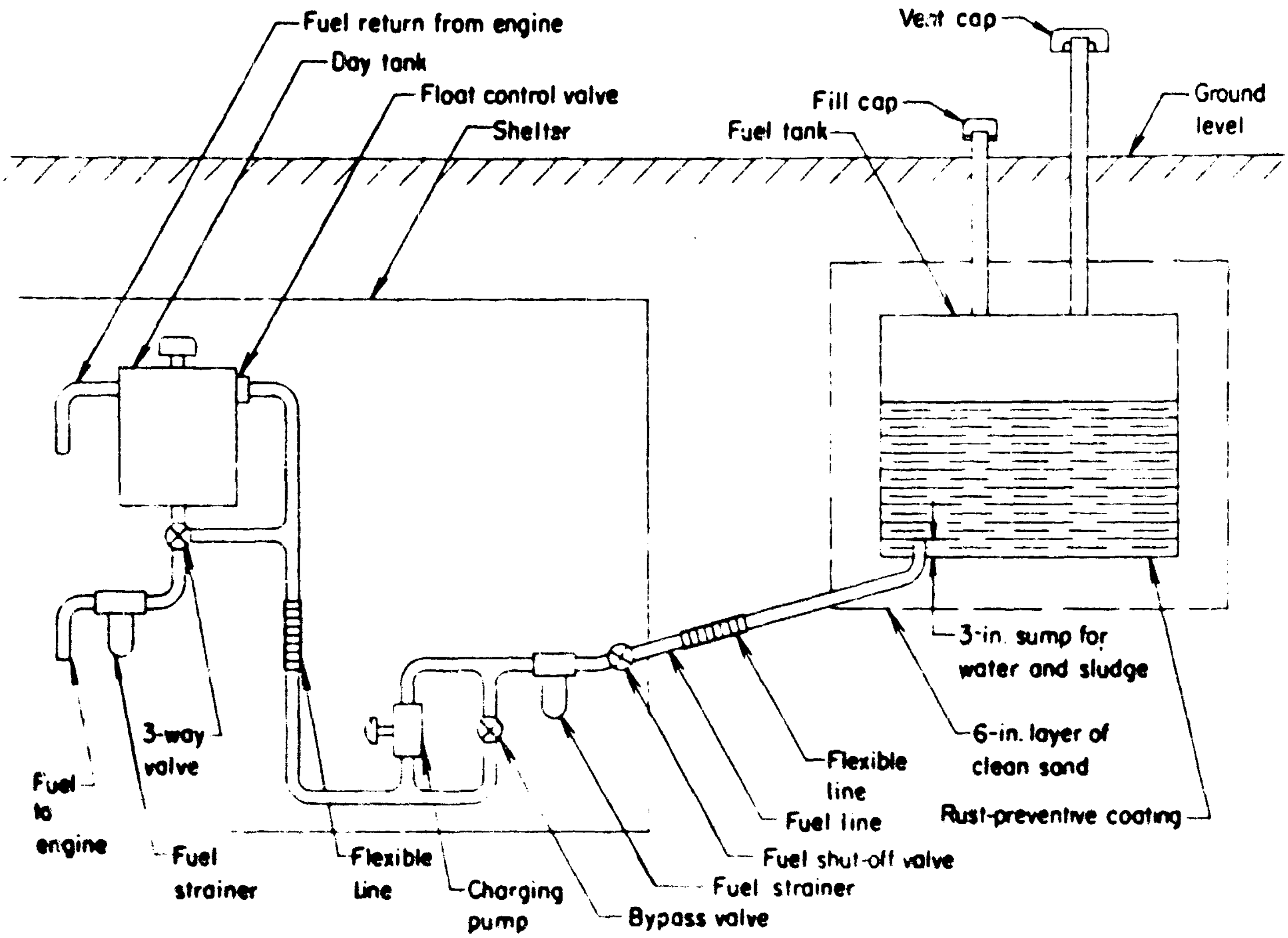


FIGURE 38. UNDERGROUND VENTED FUEL-STORAGE SYSTEM

regardless of the amount of fuel in the storage tank and of the orientation of the storage tank with respect to the prime mover.

The approximate storage life that can be expected when commercial fuels are stored in vented tanks was presented in Table 6. The data of this table indicate that the potential storage life of all of the fuels is approximately double when a vented tank is located underground rather than aboveground. The additional advantages of the underground tank (i.e., better blast and radiation protection, reduced danger from weathering, falling debris, and tampering, and reduced temperature extremes) are reasonable justification for installing the fuel storage tank underground.

If the fuel is to be completely replaced periodically and often enough to guarantee acceptable fuel quality at any time, the replacement intervals must be about half the storage times shown in Table 6 due to the uncertainties in the storage life data. The old fuel which is removed from the storage tank should have a resale value fairly close to its original value; hence the cost of the fuel

replacement program would be largely the labor involved in pumping out the old and pumping in the new fuel. The frequent-replacement approach, assuming a properly chosen replacement interval, should be reliable, and at the same time make unnecessary repeated laboratory analysis of the stored fuel.

Long-Term Fuel Storage

The data of Table 6 show that fuel storage life can be increased by sealing the underground tank. Sealing the tank will limit the amount of moisture and oxygen to which the stored fuel is exposed and will also limit the rate of evaporation. To change a vented storage system, as shown in Figure 38, to a partially sealed system requires only replacing the vent cap with pressure and vacuum relief valves which are designed to seal at pressures up to 2 psig and to admit air at a vacuum of about 1/16 psig. If the proper ratio between air space and fuel volume is used, valves of this type will reduce tank breathing considerably. Commercial fuel tanks are generally pressure tested at 5 psig; consequently, 2 psig would not be an excessive pressure. In some instances, fire regulations may require the addition of a safety relief valve with a pressure setting of about 2-1/2 psig. A further refinement of the partially sealed fuel storage tank system would be the addition of a dessicantcanister a few feet above ground level in the vent line. If this is done, even the small amount of air entering the air-vapor space above the fuel would contain only a negligible amount of moisture.

Figure 39 is a schematic illustration of an underground sealed fuel-storage tank with positive nitrogen pressure above the fuel. This type of fuel-storage system is frequently referred to as a nitrogen-blanket system. The nitrogen blanket prevents air and water vapor from coming into contact with the fuel. A minimum nitrogen pressure of about 0.5 psig should be maintained on the fuel, and an allowance should be made for expansion of the fuel due to seasonal temperature variations by filling the tank to only 90 or 95 per cent of its capacity. The tank must be equipped with a pressure relief valve to prevent the internal pressure from exceeding the design limit. A vacuum relief valve is required to prevent damage to the tank in case the nitrogen supply should be exhausted and subsequently the liquid level in the tank would be lowered by a decrease in temperature or by removal of fuel from the tank.

The storage system can be checked for serious leaks by simple observation of the rate of pressure loss from the nitrogen tank. Small leaks, however, may be masked by pressure fluctuations resulting from temperature changes. A small surge chamber located between the nitrogen tank and the pressure reducer will be a useful aid in determining the presence of small leaks. All but the most minute leaks can be detected if the tank valve is closed and the rate of pressure decline in the surge chamber is observed over a 10-minute period. The nitrogen consumption of such a system would have to be determined after installation. Because pressurized nitrogen-blanket storage systems are seldom used, such an installation might require special approval from state and local authorities.

Figure 40 is a schematic illustration of an underground storage tank for LPG. LPG is supplied and stored in pressure vessels as a liquid. The storage pressure varies with the ambient temperature and can approach 200 psi in warm weather. The liquid must be vaporized by heat addition before it can be passed through a carburetor and into an engine for combustion.

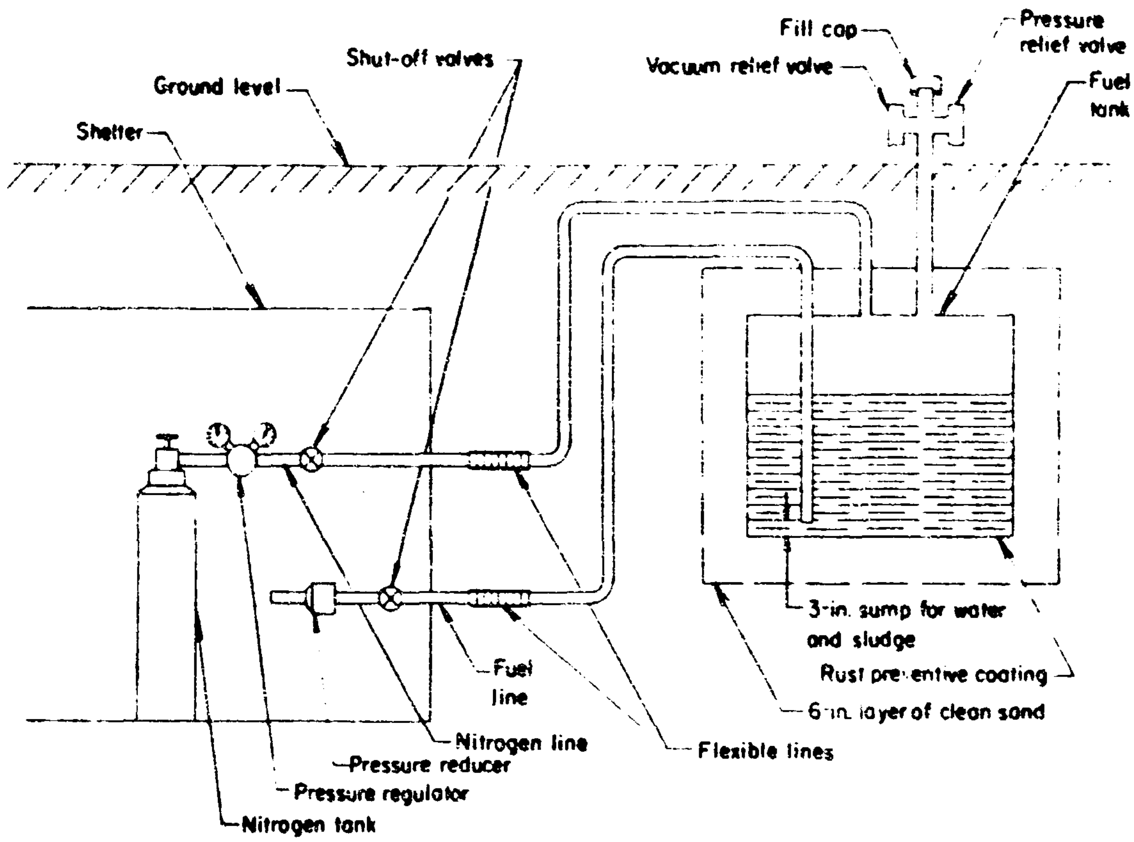


FIGURE 39. UNDERGROUND SEALED FUEL-STORAGE SYSTEM W'TH POSITIVE NITROGEN PRESSURE

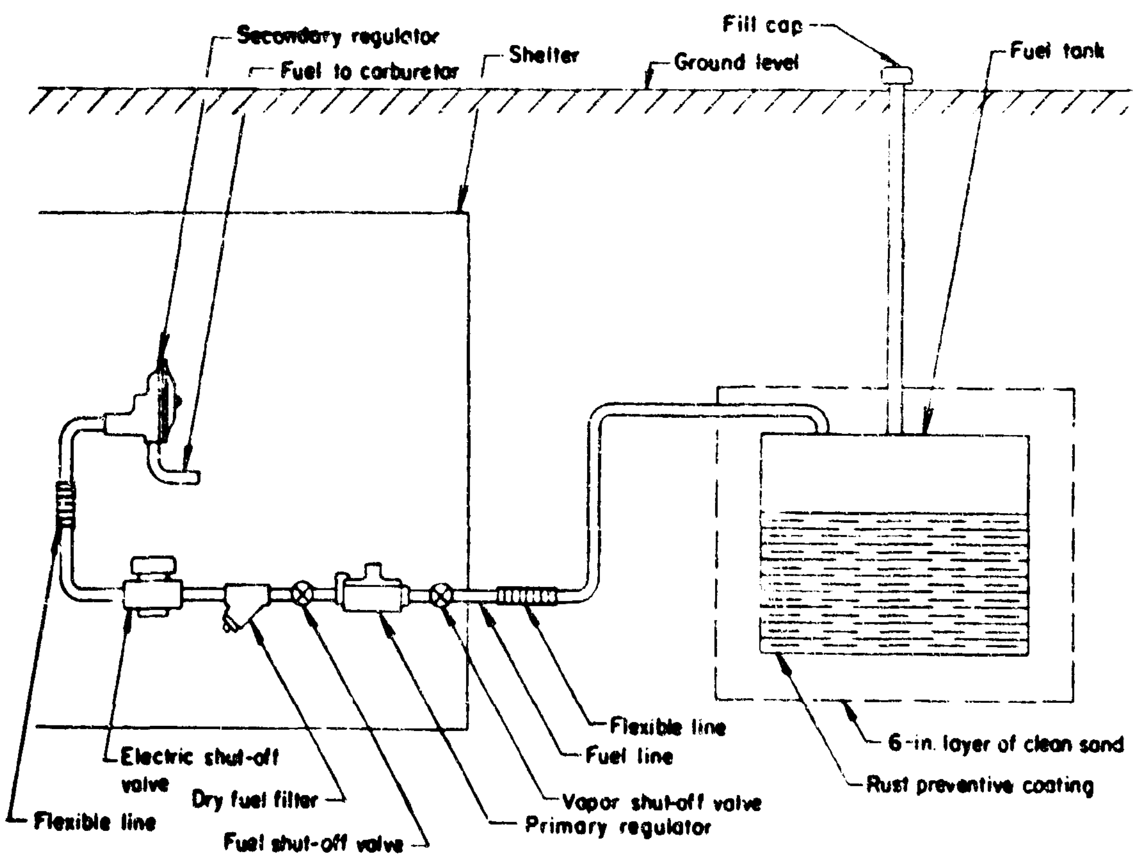


FIGURE 40. UNDERGROUND LPG FUEL-STORAGE SYSTEM

There are two methods in common use for withdrawing LPG fuel from the storage tank during engine operation. These two methods are vapor withdrawal and liquid withdrawal. The vapor withdrawal method is illustrated in Figure 40. In this system the fuel vaporizes in the tank above the liquid surface. The heat required for vaporization is supplied by the earth surrounding the tank. The vapor, which is above atmospheric pressure, flows through the fuel line, a primary regulator, a dry-type filter, an electric shut-off valve, and finally through a secondary regulator, to the carburetor. As long as there is sufficient new vapor formed in the tank to supply the engine requirements, the vapor withdrawal system will function satisfactorily.

It has been found by experience that a 500-gallon tank half full and buried at least 2 feet below the frost line will vaporize 8-1/2 gallons per hour at 40 F, and a half-full 1000-gallon tank under the same conditions will vaporize 15 gallons per hour. A 100-hp prime mover operating at full load will consume about 12 gallons per hour and will require a total fuel supply of about 4000 gallons for a 2-week emergency period. If this fuel were stored in a 5000-gallon tank buried at least 2 feet below the frost line, the vaporization rate should be more than adequate.

The liquid withdrawal method requires an artificial source of heat for vaporization of the fuel. The fuel is piped in liquid form to the engine and passed through a vaporizer unit which includes both primary and secondary regulators. The usual source of heat for the vaporizer is hot water from the engine jacket; however, a problem arises in starting and warming up the engine before sufficient jacket heat is available. Smaller engines can frequently be started and warmed up on the vapor which naturally forms in the tank above the liquid. This would depend on the volume of the space above the liquid and on the ambient temperature. For maximum reliability an external source of heat must be provided for starting and warming up the engine. A battery-powered electric heating element or a home "handyman"-type propane torch could be used to provide the necessary emergency heat.

As shown in Table 6, the estimated storage life of LPG is quite long compared with that of other petroleum-based fuels. The LPG tank is sealed and pressurized, and if it is properly purged and filled initially, there should be no air or water in the system.

LPG is heavier than air and thus will seek and settle in low places. Positive means must be provided to sweep out any leakage to avoid any explosion hazard. Numerous codes and regulations govern the installation of LPG fuel systems, and these should be followed rigidly.

Fuel Performance Characteristics

Table 7 lists the common petroleum-based fuels considered in this study with representative values for physical and performance characteristics. The cost figures were obtained from local suppliers in the Columbus, Ohio, area. The fuel-rate data were obtained from the fuel consumption curves presented in the Basic Prime Movers section of this report. The least expensive fuel and also the fuel requiring the least storage volume is No. 2 fuel oil. Gasoline is the most expensive fuel, being twice as costly for a 2-week period as No. 2 fuel oil, and LPG requires the greatest storage volume, almost twice that required for No. 2 fuel oil.

TABLE 7. REPRESENTATIVE PHYSICAL AND PERFORMANCE CHARACTERISTICS FOR THE COMMON FUELS

Fuel	Weight, lb/gal	Lower Heating Value, Btu/lb	Cost, $/gal	100-hp Prime Mover: Fuel Rate(a) lb/hr	100-hp Prime Mover: 2-Week Supply(b) gal	100-hp Prime Mover: 2-Week Supply(b) $
Gasoline	6.07	18,750	0.179	63	3490	625
Kerosene	6.80	17,550	0.160	44	2170	347
No. 2 Fuel Oil	7.05	17,992	0.150	43	2050	307
LPG	4.24	20,015	0.135	51	4040	545

(a) Assuming gasoline used in gasoline engine, kerosene and No. 2 fuel oil used in turbocharged diesel engine, and LPG used in LPG engine.
(b) Assuming full-load operation for full two weeks.

• WASTE-HEAT RECOVERY

- WASTE-HEAT AVAILABILITY
- WASTE-HEAT UTILIZATION
- WASTE-HEAT RECOVERY SYSTEMS
- WASTE-HEAT RECOVERY TESTS

WASTE-HEAT RECOVERY

Conventional prime movers convert only 15 to 35 per cent of the fuel energy supplied to useful shaft work. The bulk of the remaining energy is rejected as heat to a cooling system or in exhaust gas discharged to the atmosphere. A small percentage of energy is rejected as radiation and in miscellaneous other ways. Designers and users of combustion engines have long sought practical means to recover and use this rejected heat energy and they have had some degree of success. Heat exchangers are commercially available for the recovery of engine cooling water heat and exhaust gas heat for use in space heating, water heating, and production of low pressure steam.

Community shelters will require a certain amount of total power in various energy forms. Among the possible energy requirements are: electrical energy for lighting and communications; mechanical energy for ventilation, air conditioning, and water circulation or pumping; and heat energy for water heating, air conditioning, and cooking. The total energy requirement will determine the size of the prime mover to be provided. However, the size of the prime mover for a given shelter could be significantly reduced if a substantial portion of the energy rejected from the auxiliary power system could be recovered and put to use. Thus, the primary power output of the prime mover, i.e., the shaft power, could be used to supply the mechanical energy requirements of the shelter; the recovered energy could be used to supply the heat energy requirements of the shelter. It is also conceivable that, if the heat energy requirements are relatively small, some of the recovered heat energy could be converted to assist the prime mover in meeting the mechanical energy requirements. This section of the report deals with the potential for recovering waste heat from the cooling and exhaust systems of the prime mover. Means for utilizing recovered waste heat in the shelter are discussed and specific means for recovering waste heat are described. The results of laboratory tests with simple exhaust heat-recovery equipment are presented.

Waste-Heat Availability

Table 8 shows typical heat balances, heat-rejection rates, and exhaust gas temperatures at continuous rated load for various commercial prime movers. Although these data are representative, there may be wide variations between actual engines of the same general type. The heat balance shows how the engine utilizes or rejects the fuel energy supplied. Regardless of engine type, the major portion of the fuel energy is rejected as heat, which is transferred to the coolant, carried away in the exhaust gases, or radiated to the surroundings. The percentage actually converted to shaft power varies from 15 to 35 per cent depending on the type of prime mover. The coolant and exhaust heat-rejection rates represent the total amount of heat that could be recovered under ideal conditions from the cooling and exhaust systems. The exhaust gas temperatures are indicative of the potential energy level of the exhaust heat energy.

For practical purposes all of the heat rejected to the coolant is recoverable with a properly designed cooling system. The temperature level of this recoverable heat is relatively low, about 200 F at the most. Therefore, the energy level is quite adequate for space or water heating but inadequate for conversion to mechanical shaft power. Only about 60 to 80 per cent (depending on the initial exhaust gas temperature) of the heat rejected in the exhaust

TABLE 8. TYPICAL HEAT BALANCES, HEAT-REJECTION RATES, AND EXHAUST GAS TEMPERATURES

Performance Parameters	Four-Cycle Spark Ignition	Two-Cycle Diesel	Four-Cycle Diesel		Gas Turbine	
			TC(a)	NA(b)	NR(c)	R(d)
Fuel energy converted to power, per cent	26	30	35	31	15	25
Fuel energy rejected to coolant, per cent	30	21	22	26	--	--
Fuel energy rejected in exhaust, per cent	32	37	29	30	70	65
Fuel energy rejected as radiation, per cent	12	12	14	13	15	10
Coolant heat loss, Btu/hp-hr	2,900	1,800	1,600	2,100	--	--
Exhaust heat loss, Btu/hp-hr	3,100	3,100	2,100	2,500	11,900	6,600
Exhaust gas temp, F	1,200	600	800	900	1,000	500

(a) Turbocharged
(b) Naturally aspirated
(c) Nonregenerative
(d) Regenerative

gases is recoverable for practical purposes. The reason for this limitation is that the exhaust gas cannot be cooled below about 300 F without risk of serious corrosion problems in the exhaust system because of the potential condensation of highly corrosive constituents in the exhaust gases. The energy level of recoverable exhaust heat is significantly higher than that of recoverable cooling-system heat and, therefore, this energy may be used to produce shaft power as well as for space and water heating.

Waste-Heat Utilization

Waste-heat recovery in commercial power installations is becoming more prevalent.[13,14] In fact, waste-heat recovery is a key factor in the total-energy-package concept which is being promoted at the present time particularly

for shopping centers, apartment house projects, and schools. The primary justification for waste-heat-recovery or total-energy-package systems is operating economy. The concept is generally sold where it can be demonstrated that the higher first cost of the equipment will easily be offset by the lower fuel and maintenance costs during the life of the installations.

In an application such as the community shelter system where the operating life of the equipment is measured in hundreds of hours rather than tens of thousands of hours, it would appear that waste-heat recovery could not easily be justified on the basis of operational economy. The problem of determining feasibility can be somewhat simplified by dividing it into two approaches: (1) recovery of waste heat for conversion to auxiliary shaft power and (2) recovery of waste heat for heating such as space and water.

Several types of systems for conversion of waste heat to power were evaluated. These were: Brayton-cycle, Stirling-cycle, and Rankine-cycle external combustion engines; blowdown turbine direct mechanical energy conversion; and thermoelectric and thermionic solid-state direct electrical energy conversion. The Brayton, Stirling, and Rankine cycles would utilize the waste heat directly from the exhaust gases to supply heat to an internal working fluid; air for the Brayton and Stirling cycles, and water for the Rankine cycle. The blowdown turbine would utilize the kinetic energy of the exhaust gases. The thermoelectric and thermionic solid-state devices would utilize the waste heat directly from the exhaust gases to develop electrical energy.

Table 9 summarizes the results of the evaluation of these various energy-conversion systems. As is shown in the table, all of the energy-conversion systems except possibly the Rankine (steam) cycle are not suitable for utilizing prime-mover waste heat for one or more reasons; equipment not commercially available, equipment very expensive, waste heat at a low temperature level, low operating efficiency. Because of the relatively low temperature of the heat source, i.e., the exhaust gases, all of the possible energy-conversion systems would operate at low efficiency.

Because steam system equipment is commercially available, an analysis was made to determine the approximate economics of an actual steam-power waste-heat recovery system. If, for example, a shelter has a total power requirement for 100 hp, a satisfactory single four-cycle naturally aspirated diesel engine can be obtained for approximately $50 to $60 per hp. A complete waste-heat recovery system for this engine may recover the equivalent of about 150 hp in heat energy. If this heat energy is utilized in a steam-power system the overall efficiency of the steam-power system is likely to be only about 10 per cent for the size range. Thus, with full utilization of the engine waste heat only about 15 hp will be developed. Then the prime mover size could be reduced resulting in a saving of about $700 on the purchase price. However, a commercial exhaust heat exchanger for this system would cost about $1200; and it is likely that supplying the steam engine (or turbine), condenser, and other necessary components for the steam-power system would easily double that cost. A total investment of over $2500 would thus be required to save $700 on the cost of the prime mover.

In addition to the saving of $700, a saving in fuel and fuel storage costs and savings in the first costs of auxiliary systems such as the starting system and cooling system must also be considered. However, these savings are not likely to total $1800.

TABLE 9. COMPARISON OF ENERGY CONVERSION SYSTEMS FOR UTILIZATION OF PRIME-MOVER WASTE HEAT

System	Equipment Available	Probable Cost	Complexity	Remarks
External combustion engines				
Brayton cycle	No	High	Medium	
Stirling cycle	No	High	High	Not self-starting. Potentially low per cent heat recovery.
Rankine cycle	Yes	Medium	Medium-high	Requires supply of fresh water.
Solid-state devices				
Thermoelectric	Yes	Very high	Low	Availability limited. Estimated cost $2500 per kw.
Thermionic	No	High	Medium-low	Requires high temperature heat source.
Direct mechanical				
Blowdown turbine	No	High	Medium	Requires extensive modifications to engine.

In addition to the lack of economic justification for a waste-heat recovery system of this type there would be a number of disadvantages. a reduction in the total system reliability because of the addition of complex equipment, an increase in the installation and maintenance costs and problems, and an increase in the shelter heat-rejection load because of the inefficiency of the steam system. Therefore, recovery of waste heat for conversion to power is not feasible.

The second approach to waste-heat recovery for the shelter system (water or space heating) can be considered practical and justified if it is assumed that there will be a demand for hot water and/or for heat in the occupied space in the shelter. If there is such a demand and if it can be met with low energy hot water or steam, the simplest approach would probably be to use a heat exchanger or an ebullient-type cooling system and to draw off the required amount of hot water or steam as it is needed. The cooling system in this case should be large enough to reject the entire prime mover cooling load to the heat sink when necessary. It is quite possible that waste-heat recovery using only the jacket water would be adequate to service the space heating and hot water needs of most shelters; inasmuch as the heat rejected to the coolant is not less than 50 per cent and in one case greater than 100 per cent of the shaft power output (see

Table 8). This means that for a 100-hp prime mover the amount of cooling system waste heat that could be made available to the shelter would be between 120,000 and 290,000 Btu/hr depending on the type of prime mover used.

There is an exception, of course, in the case of the gas turbine where exhaust waste-heat recovery is the only possibility. The gas turbine appears to be competing quite favorably in the present market for very large, total-energy-package systems, and it is not inconceivable that in the near future gas-turbine total-energy-package systems may be approaching a competitive status in the size range of interest in the community shelter program. Because of the present lack of commercial equipment and reliable data, however, it is not possible to make even an approximate economic analysis nor to predict when the smaller size units will become commercially available and truly competitive.

Waste-Heat Recovery Systems

As mentioned previously, waste heat is available in both the jacket water and exhaust gases of piston engines and in the exhaust gases of gas turbines. Because of the low temperature of jacket water the waste heat from this source can be used only for heating water or tempering ventilating air. Exhaust gases being at a much higher temperature can be used to produce either steam or hot water. Waste-heat-recovery systems can take many forms depending on the type and quantity of heat energy required. Three possible heat-recovery systems are discussed in the following prargraphs to illustrate equipment requirements and system capabilities and limitations. The three are: (1) conventional jacket water and exhaust-heat recovery, (2) ebullient cooling and exhaust-heat recovery, and (3) gas-turbine heat recovery.

Conventional Jacket Water and Exhaust Heat Recovery

Figure 41 is a schematic illustration of a combined jacket water and exhaust heat-recovery system. The basic components of such a system are: jacket water heat exchanger, exhaust heat exchanger, circulating pump in the heat-recovery system, control valves, and appropriate heat sinks. The system as shown is capable of producing hot water and steam in varying amounts according to the demand. The steam could be either at atmospheric pressure for heating or at an elevated pressure for shaft power production. A flow of water must be maintained through the system at all times to ensure that the engine will be adequately cooled and that the exhaust heat exchanger will not overheat and burn out. If the supply of hot water or steam exceeds the demand, the excess is diverted to the heat sink to be cooled and subsequently returned to the system by the circulating pump. If the steam pressure is significantly above atmospheric pressure, an expansion valve can be used to protect the heat exchanger, pump, etc., from overpressure.

The waste heat recoverable from a 100-hp, four-cycle, turbocharged diesel engine, using the heat-recovery system illustrated in Figure 41, and the heat-rejection rate data in Table 8, would be approximately 305,000 Btu/hr. With this amount of heat, approximately 365 gal of water could be heated through a 100 F temperature rise. About 47 per cent of this heat would be recovered from the exhaust system using an exhaust gas lower temperature limit of 300 F. If a smaller quantity of hot water were required, the heat-recovery system could

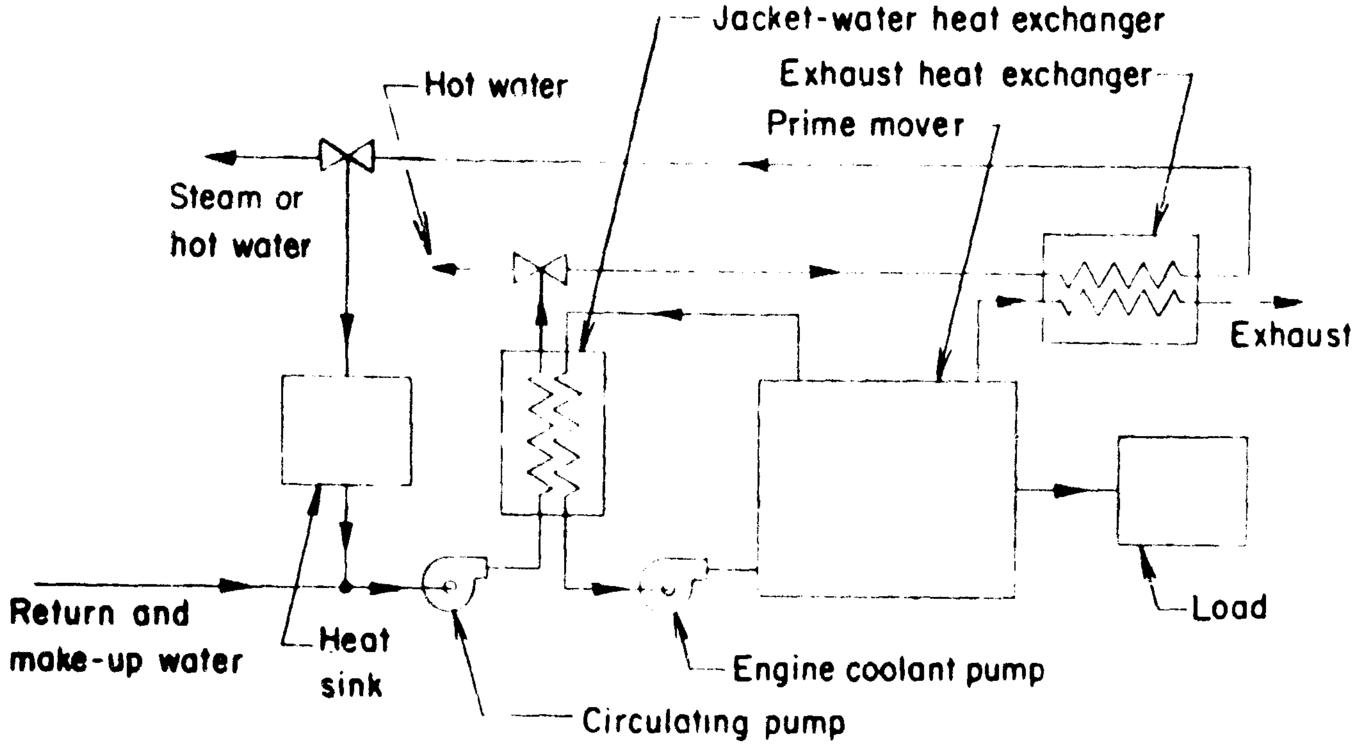


FIGURE 41. JACKET WATER AND EXHAUST WASTE-HEAT RECOVERY SYSTEM

be simplified by installing only one of the two heat exchangers realizing that if steam were required the exhaust-gas heat exchanger would have to be used. As with the combined system, provision would have to be made to insure that with a jacket water system the engine would be adequately cooled at all times. Also, with an exhaust system excess system pressure and heat-exchanger temperature would have to be avoided.

Ebullient Cooling and Exhaust Heat Recovery

Figure 42 is a schematic illustration of an ebullient cooling and exhaust heat-recovery system. The ebullient cooling system is, in itself, a producer of low-pressure steam. In the system shown in Figure 42 the steam output is increased by circulating the cooling water from the steam separator tank through an exhaust heat exchanger. Hot water can be taken for use directly from the steam-separator tank. As in the system illustrated in Figure 41, a heat exchanger and heat sink must be provided for disposal of excess recovered heat.

The orientation of the steam separator and the exhaust heat exchanger as shown in Figure 42, will assure a continuous flow of water through the exhaust heat exchanger by convection. If a sufficient supply of make-up water is available, the waste-heat recovery systems shown in Figures 41 and 42 could both be operated without heat sinks. In this case the steam and hot water in excess of shelter demand would be rejected outside of the shelter.

The waste heat recoverable with the heat-recovery system illustrated in Figure 42 would be exactly the same as that for the system illustrated in

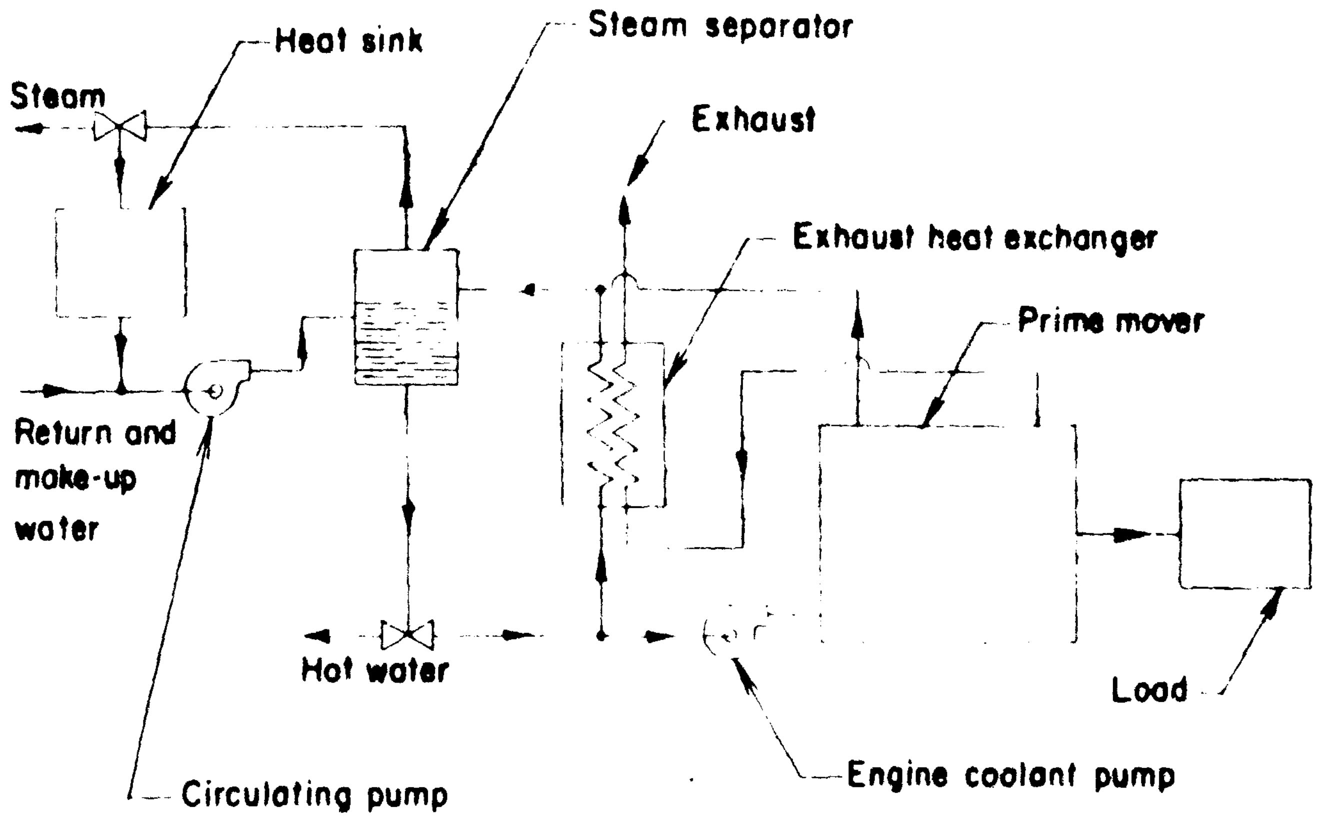


FIGURE 42. EBULLIENT COOLING AND EXHAUST WASTE-HEAT RECOVERY SYSTEM

Figure 41. For the 100-hp, four-cycle, turbocharged diesel engine example, the total heat recovered would be 305,000 Btu/hr.

Gas-Turbine Exhaust-Heat Recovery

Figure 43 is a schematic illustration of a gas-turbine heat-recovery system. The gas turbine is unique among the conventional prime movers included in this study in that it has no cooling system, all of its waste heat being rejected in the exhaust gases. Consequently, the only means of recovering waste heat from a gas-turbine prime mover is with an exhaust heat exchanger. The recovered heat, as shown in Figure 43, could be used to power an absorption-refrigeration air-conditioning system, and/or to provide hot water for heating, etc.

The energy available in the exhaust of a gas turbine can be increased over a useful range of values independently of the shaft power output of the turbine by by-passing the combustion products around the power turbine and by use of an auxiliary combustor. The energy available for recovery in the cooling and exhaust systems of diesel, gasoline, and LPG engines varies almost directly with the shaft load. Thus, the gas turbine has a significant advantage in adaptability to a waste-heat recovery system where mechanical and heat loads may vary proportionately to each other.

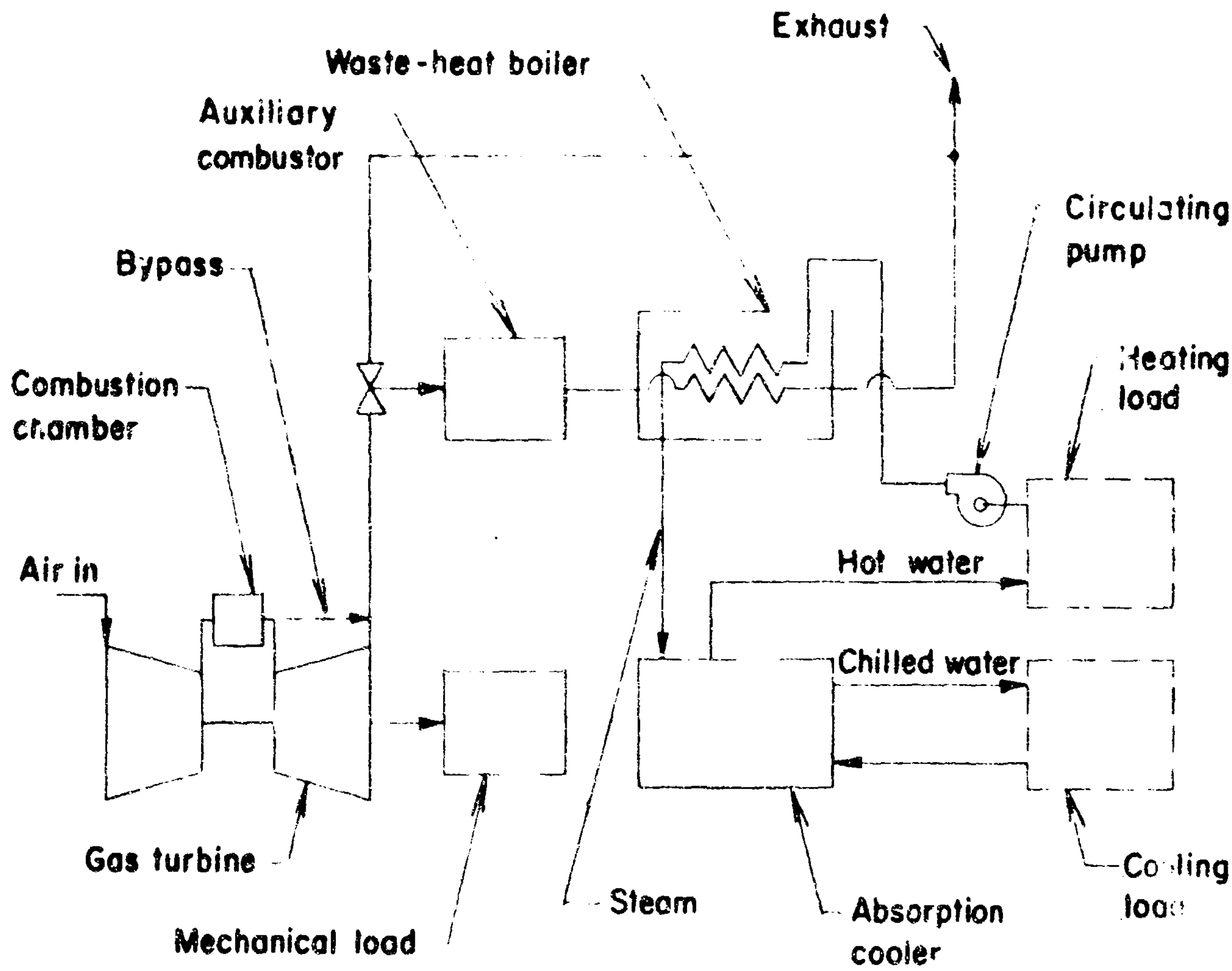


FIGURE 43. GAS-TURBINE WASTE-HEAT RECOVERY SYSTEM

The waste heat recoverable from a 100-hp regenerative gas turbine, using the heat-recovery system illustrated in Figure 43 and the data of Table 8, would be approximately 315,000 Btu/hr which would produce 380 gal of water through a 100 F temperature rise. As for the piston engines, the exhaust gas lower temperature limit was assumed to be 300 F.

Using the same assumptions as above, a nonregenerative gas turbine of 100-hp capacity would have 830,000 Btu/hr of recoverable waste heat in the exhaust gases. With this amount of heat, approximately 1,000 gal of water could be heated through a 100 F temperature rise.

Waste-Heat Recovery Tests

Jacket water waste-heat recovery systems would utilize water-to-water heat exchangers which are readily available in a large variety of types and sizes at a reasonable cost. However, commercial exhaust-heat recovery equipment is not readily available particularly for small engines because of a lack of demand, and

is very expensive because of the long design life. For these reasons, laboratory tests were conducted to determine the performance of simple and inexpensive exhaust waste-heat recovery equipment.

The 20-kw diesel engine-generator set used as a demonstration unit was used for the exhaust waste-heat recovery tests. Four simple heat-exchanger designs were evaluated. These were: (1) straight tube, (2) coiled tube, (3) baffled tube, and (4) straight-fin tube. The heat recovered with the straight tube was about half of that recovered with the other three configurations.

Figure 44 shows the laboratory setup used to evaluate the various heat-exchanger designs.

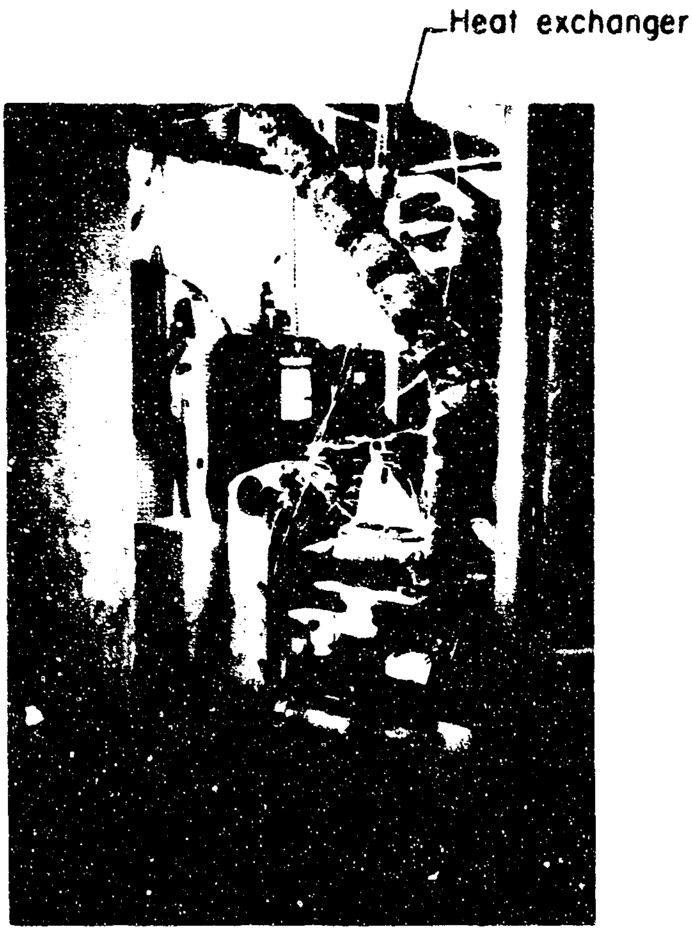
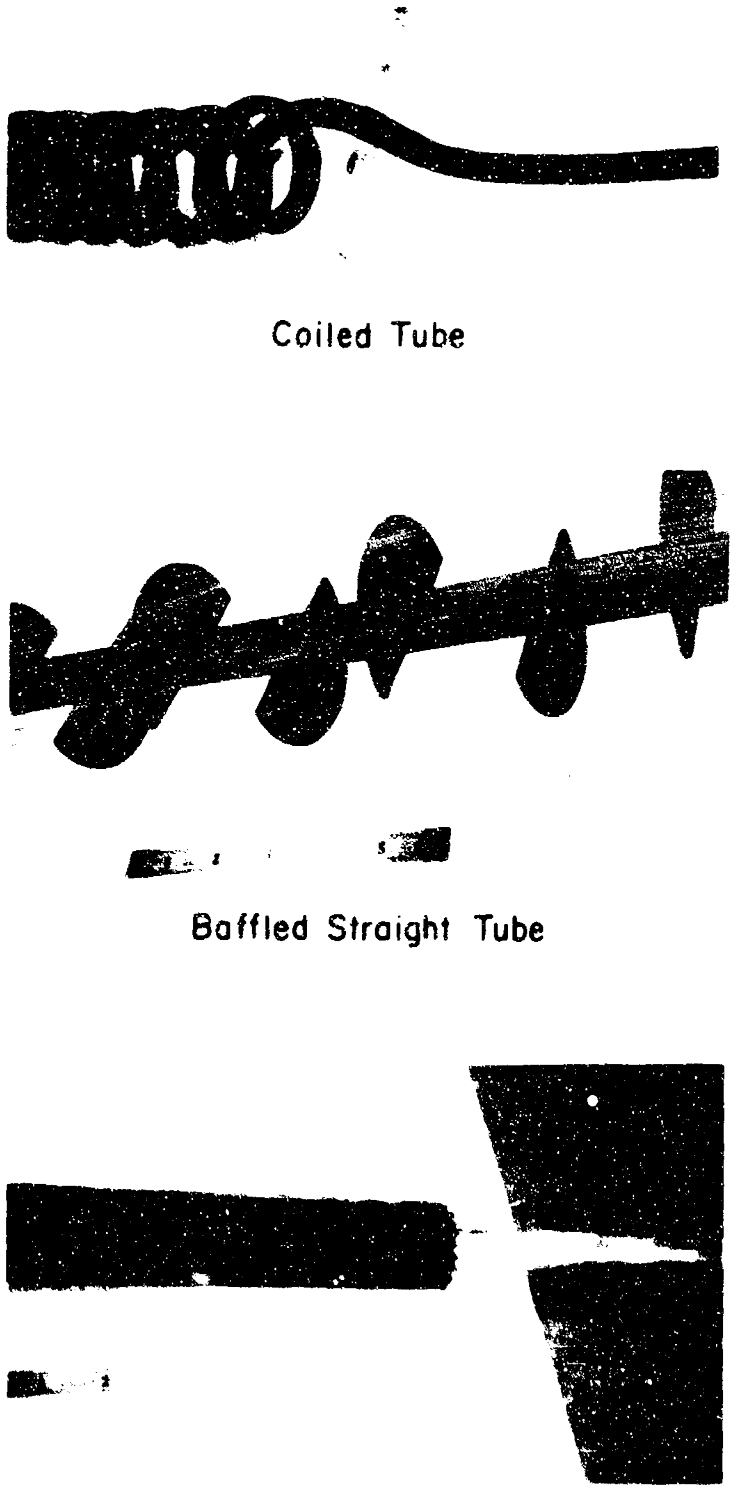


FIGURE 44. LABORATORY EXHAUST HEAT-RECOVERY TEST SETUP

Figure 45 shows three of the four exhaust heat-recovery heat exchangers tested. The coiled-tube heat exchanger was an 18-ft length of 3/8-in. copper tubing wound into a continuous coil having an inside diameter of 1-1/2 in. and a length of 3 ft which was constructed at an estimated cost of $10. The baffled straight-tube heat exchanger was fabricated by soldering 3-in. radius, semi-circular brass baffles to a 3/4-in. Admiralty brass tube at intervals of 2 in. over a 3-ft length. This configuration cost an estimated $40 to construct. The straight-fin tube heat exchanger was purchased already fabricated for $50.



Coiled Tube

Baffled Straight Tube

Finned Straight Tube

FIGURE 45. HEAT EXCHANGERS FOR EXHAUST HEAT-RECOVERY TESTS

It contained 20 radial fins, each 3-ft long, 1/2-in. high, and 0.025-in. thick, soldered to a 3/4-in. Admiralty brass tube. A fourth heat exchanger, not illustrated in Figure 45, was a straight 3-ft length of 3/4-in. Admiralty brass tube without fins or baffles which was purchased for $6. Admiralty brass was selected for the straight-tube heat exchangers because of its resistance to corrosion by exhaust gases. Copper tubing was used for the coil heat exchanger because of its easy workability and availability.

For the tests, each heat exchanger was installed, in turn, in a 3-ft section of insulated 3-in. pipe through which the exhaust gases passed. This exhaust system test section was identified in Figure 44. Both exhaust gas and cooling water temperature measurements were made at each end of the test section. Exhaust back pressure was also measured.

The coiled-tube heat exchanger was tested with supply water circulated directly through it and discharged to a drain. The three straight-tube heat exchangers were tested in a free-convection system which included a 20-gal hot water reservoir open to the atmosphere. Supply water was introduced into the system about midway in the reservoir and hot water was allowed to overflow at the top to maintain a desired maximum water temperature.

Figure 46 shows the heat recovered from the exhaust gases by the different heat exchangers as a function of generator load. A sketch of the test section with the coiled tube in place shows the direction of exhaust gas and water flow for all the tests. The water and gas flows are parallel; however, calculations indicated that counter-flow operation would increase the heat transfer by only about 1 per cent.

Table 10 lists additional performance results from these tests. These data are all representative of full-load operation. The temperature of the exhaust gases in the exhaust manifold varied from 975 to 1075 F during the tests, and the back pressure in the exhaust system before the test section was installed was about 11 in. H_2O. The computer water-flow-rate data are based on a water-supply temperature of 60 F and an 80 F rise. The per cent heat recovery is based on an ambient temperature of 100 F.

A length of 3 ft was arbitrarily selected for all of the test heat exchangers for convenience and for direct comparison purposes. Longer lengths would naturally promote higher heat recovery up to the limiting minimum exhaust-gas temperature of 300 F at which condensation of exhaust gases occurs. For the short use period of 2 weeks it would probably be acceptable to design for even lower final exhaust-gas temperatures.

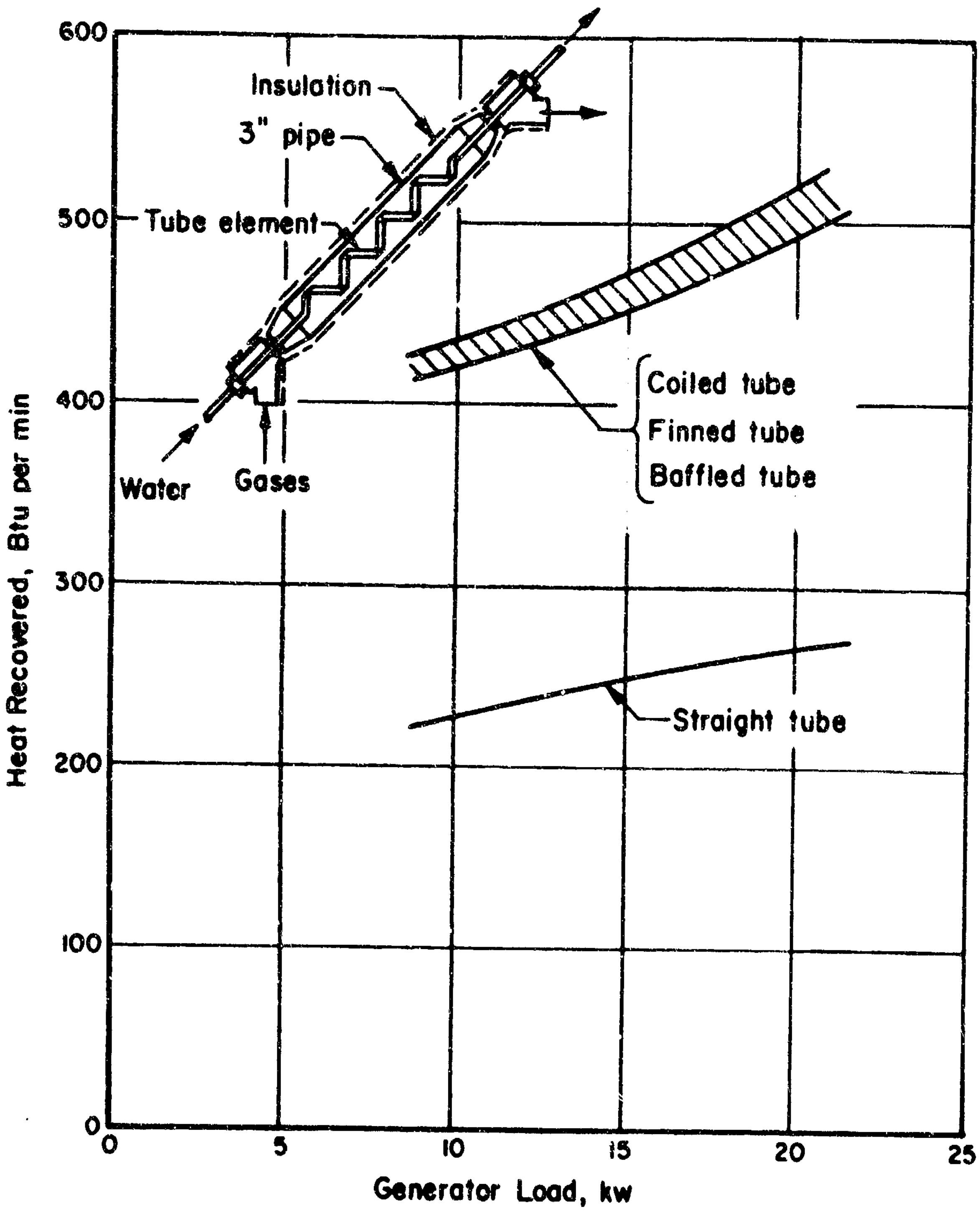


FIGURE 46. EXHAUST WASTE-HEAT EXCHANGER PERFORMANCE

TABLE 10. EXHAUST GAS HEAT-EXCHANGER PERFORMANCE AT FULL-LOAD ENGINE OPERATION

Heat Exchanger	Exhaust Gas Temperature, F Initial	Exhaust Gas Temperature, F Final	Increase in Exhaust Back Pressure in H_2O	Water Flow Rate at 80 F Temp Rise, gal per hr	Heat Recovered Btu per hr	Heat Recovered Per Cent
Coiled tube	1075	725	6	47	30,600	36
Straight tube	1015	860	5	24	15,700	17
Baffled straight tube	1075	772	59	47	30,600	31
Finned straight tube	975	665	3	45	29,500	35

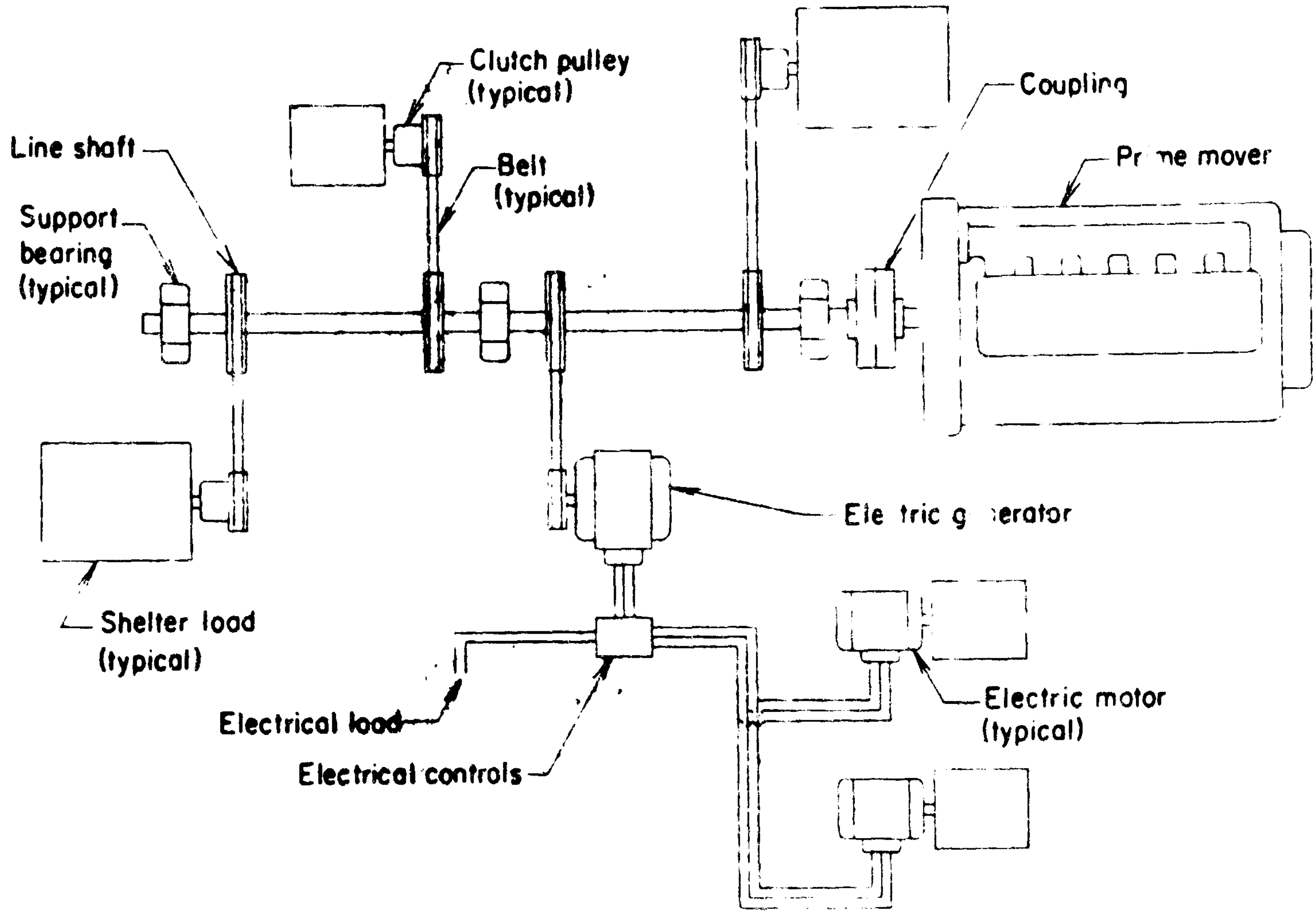


FIGURE 47. SINGLE-PRIME-MOVER MECHANICAL POWER-TRANSMISSION SYSTEM

Figure 48 shows the multiple-prime-mover approach to a mechanical power-transmission system. In this system separate smaller prime movers are provided for the two major mechanical loads and a third prime mover is coupled at one end to a small load and at the other end to an electric generator which supplies power for the purely electrical requirements and for the other smaller mechanical loads. All of the prime movers are provided with disengaging clutches when connected to a mechanical load.

Many factors must be considered when a mechanical power-transmission system is selected for a specific shelter application. Among these are: location of mechanically driven equipment in the shelter, relative cost of several small prime movers compared with cost of a single large prime mover, the cost and complexity of the intermediate transmission equipment for the single prime mover, the potential reliability of multiple prime movers, and the cost and complexity of installation of multiple prime movers.

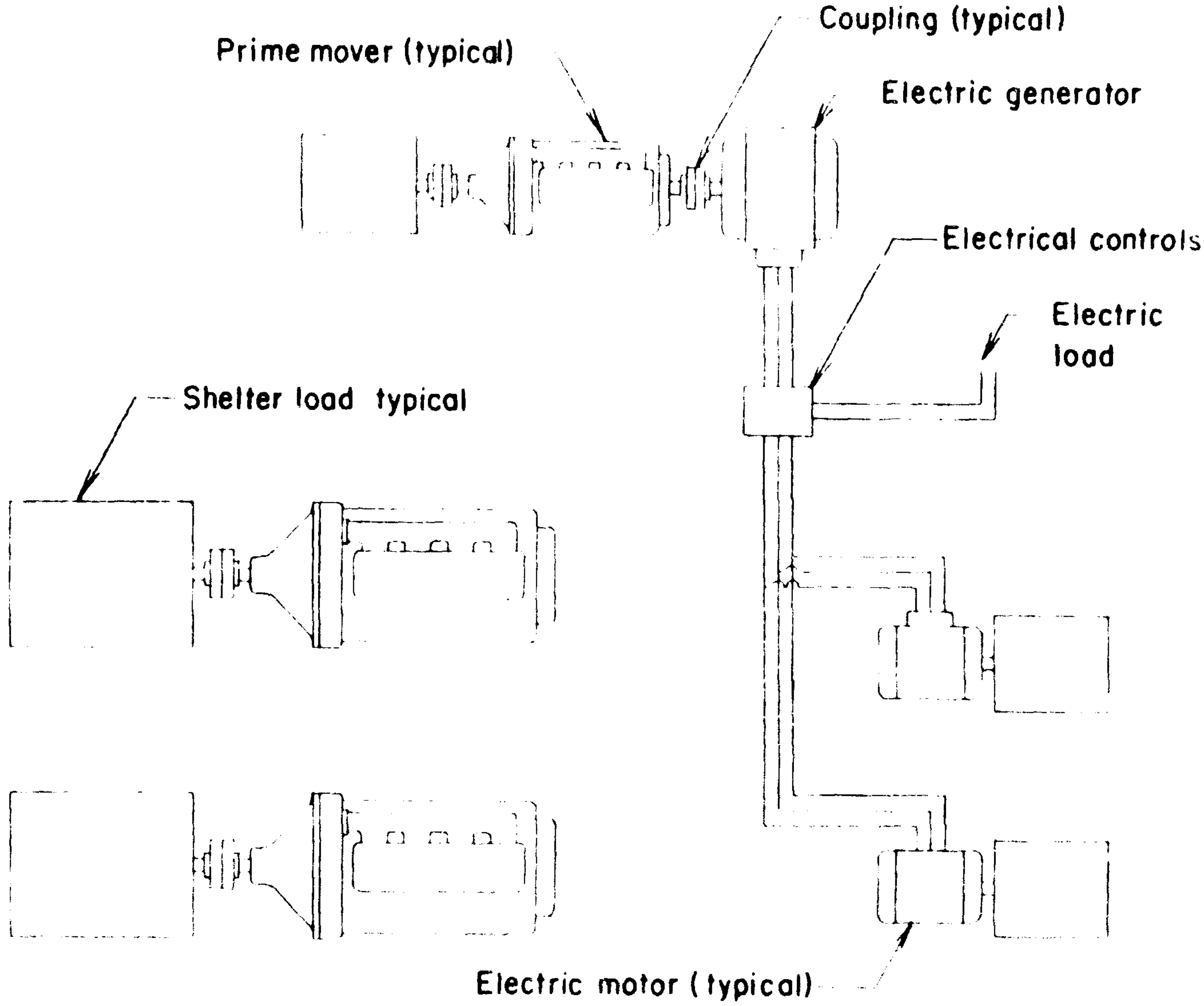


FIGURE 48. MULTIPLE-PRIME-MOVER MECHANICAL POWER-TRANSMISSION SYSTEM

Mechanical systems should have efficiencies of power transmission in excess of 90 per cent. The relatively simple direct-coupling and clutch arrangement should have an efficiency greater than 95 per cent. If gears, belts, or chains and intermediate line shafts are used, the over-all efficiency will be closer to 90 per cent. The maintenance requirements for mechanical systems are minimal. The primary considerations are lubrication of bearings and gear or chain drives if used, and protection of critical surfaces and equipment with preservative coatings. The reliability of mechanical systems in general is high. Even greater reliability can be achieved with multiple-prime-mover systems by interconnecting the starting systems so that each prime mover could provide starting power for any of the others in an emergency. If any one prime mover of a multiple system should completely fail in an emergency, partial power would still be available for the shelter; however, the equipment it was driving would be inoperative.

The components for mechanical power-transmission systems are inexpensive and readily available. The over-all cost of a single-prime-mover power-transmission system should be significantly lower than the cost of any other power-transmission system.

The multiple mechanical system would also be extremely low in cost, but it would require a relatively more expensive prime mover installation; consequently, the combined cost would probably be more comparable to that of an electric generator system.

Mechanical systems can be made acceptably safe if all moving elements such as belts, chains, gears, clutches, and couplings are suitably enclosed or guarded. Chain and gear drives are somewhat noisy. A belt drive would be significantly quieter, and would serve as a shock absorber for torque pulses between the prime mover and driven equipment, and would be more easily replaced in case of failure.

Electric-Generator Power-Transmission System

Engine-generator sets have long been the standard for emergency stand-by power systems and they are presently the most widely used. Electrical energy is easily transported from generator to motor with low transmission loss and permits practically noiseless operation. The most adaptable electric system for incorporating commercially available equipment is the four-wire, three-phase, Y-connected 120/208-volt, 60-cycle system with the generator operating at a speed of either 1200 or 1800 rpm. The generator and prime mover should utilize a common housing or be mounted on a common base to avoid troublesome misalignments.

Figure 49 is a schematic illustration of an electric-generator power-transmission system. The electrical power from the generator goes into a control box which contains voltage-control, switch gear, and circuit-breaker components. From the control box, distribution lines provide power for the various mechanical and electrical loads throughout the shelter. Commercial electrical power, if available, would also be brought into the control box and could be used for exercising operation of the shelter equipment.

Figures 50 and 51 show the full-load efficiencies of electric generators and motors, respectively. The generator efficiency curve is plotted against the electrical kilowatt output of the generator, and the motor efficiency curve is plotted against the shaft horsepower output of the motor. These data indicate that single large units are more desirable than several smaller units from the standpoint of efficiency. Some small electric motors will be needed throughout the shelter and their low efficiencies cannot be avoided. However, because small power consumptions are actually involved, their effect on the over-all efficiency of the electric-generator power-transmission system will not be great.

The efficiency of a generator or motor varies only slightly with a load change between 75 and 125 per cent of rated load. Below 75 per cent of rated load the efficiency begins to fall off and may be about 10 per cent lower at 25 per cent rated load than at full rated load. Assuming an efficiency of 91 per cent for the generator, which would be approximately correct for a 70-kw unit (about 100-hp input), and 85 per cent for an electric motor, which would be about right for a 10-hp unit (an assumed average size), the over-all efficiency would be 77 per cent. Line losses would be on the order of 1 per cent for a transmission length of 25 or 30 feet. Thus, a reasonable over-all efficiency for an electric generator power-transmission system would be about 76 per cent.

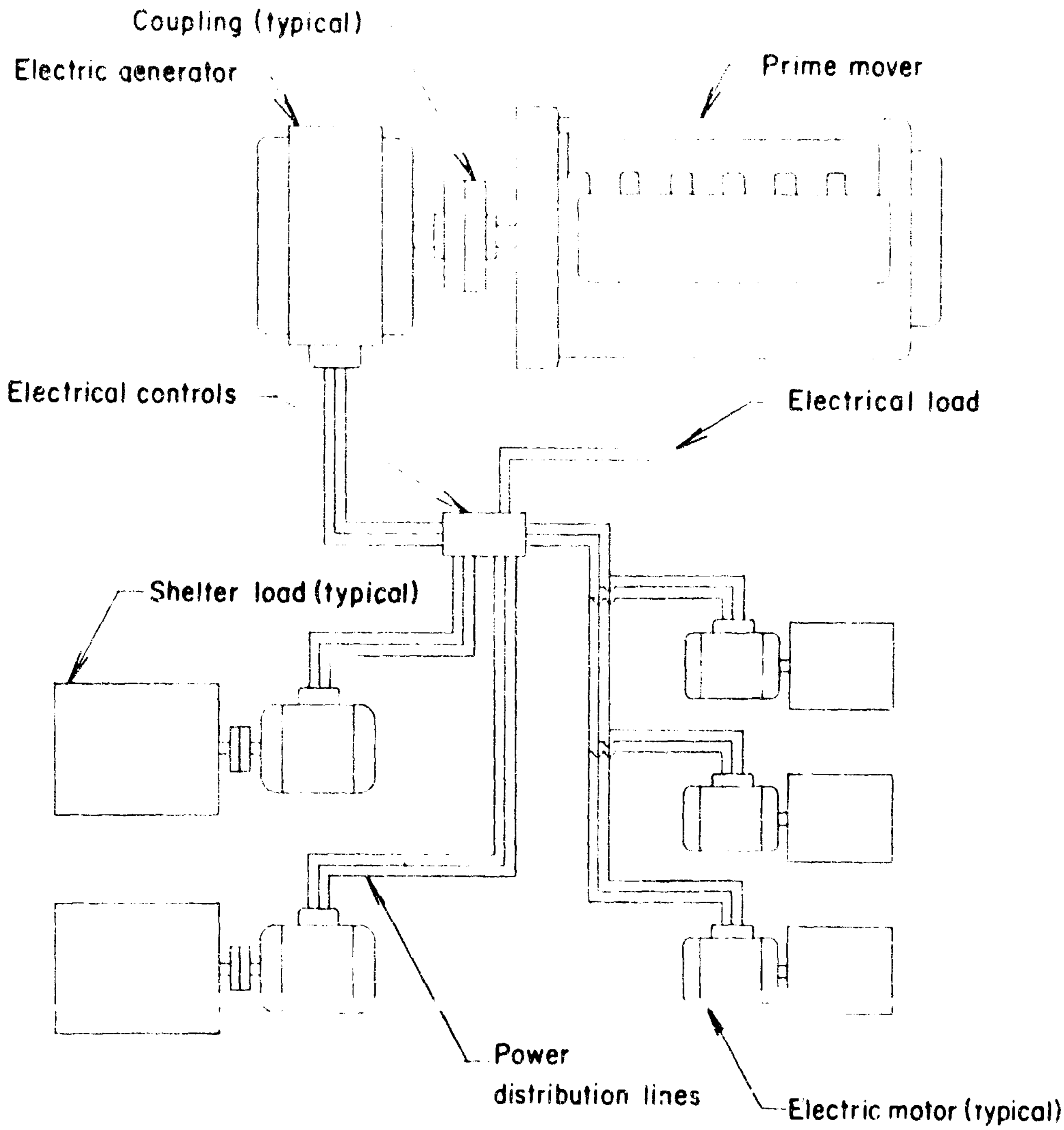


FIGURE 49. ELECTRIC-GENERATOR POWER-TRANSMISSION SYSTEM

The major problem which will be encountered in the maintenance of electric-generator power-transmission systems is the effect of moisture. The amount of moisture present in the shelter air will determine the frequency of maintenance and testing. It will also determine the type of insulating materials to be used in both the generator and the motors. Condensation of moisture in the generator or motor could cause rusting of bare iron surfaces and the formation of mold, and eventually could lead to either mechanical or electrical failure of the equipment. Methods for reducing moisture absorption in generators and motors or providing resistance to moisture include: heating to maintain equipment above the dew point, dehumidifying the entire shelter, fully enclosing the generator and motors, encapsulating the stators, and specifying materials impervious to moisture.

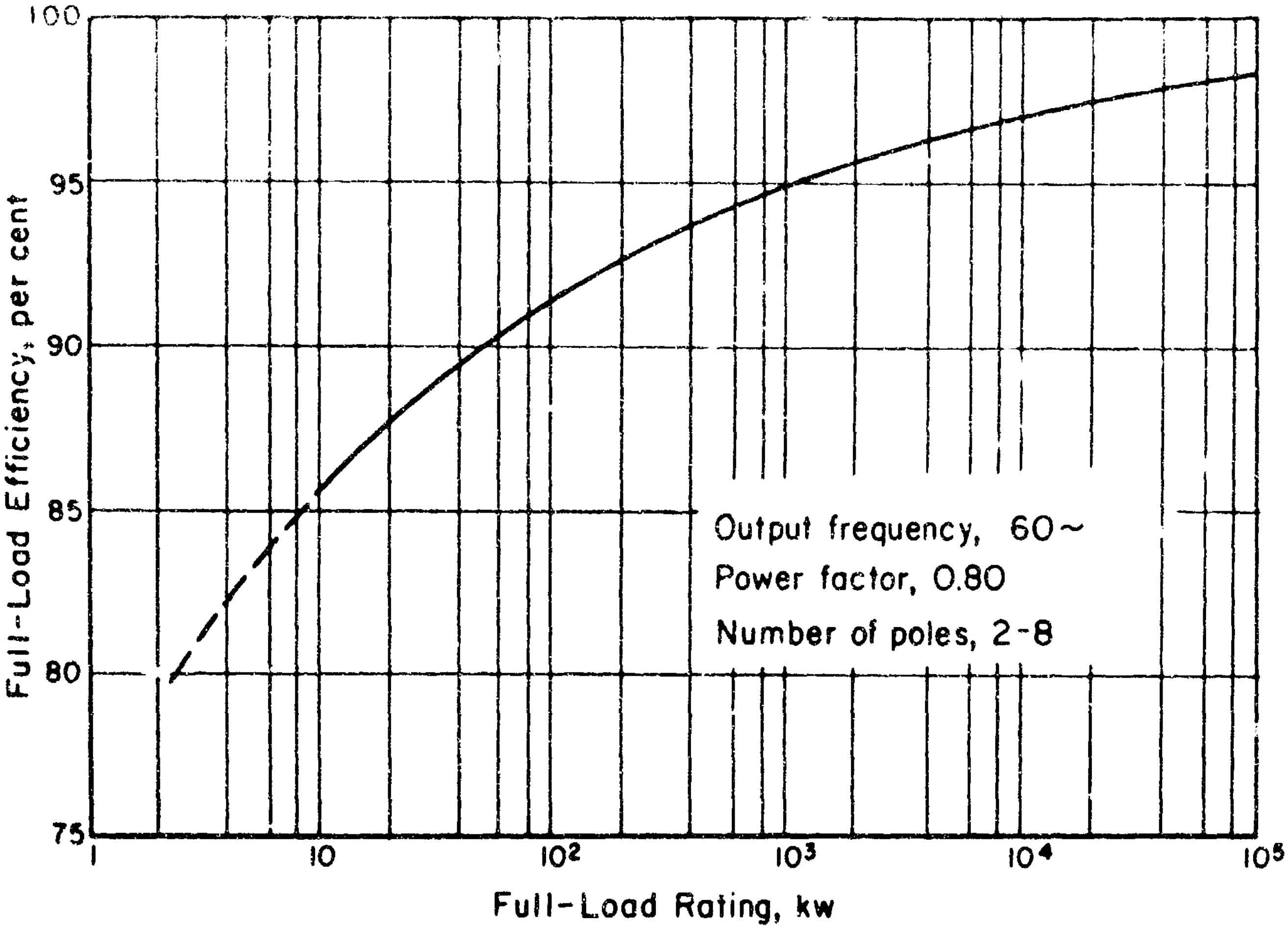


FIGURE 50. ELECTRIC GENERATOR EFFICIENCY

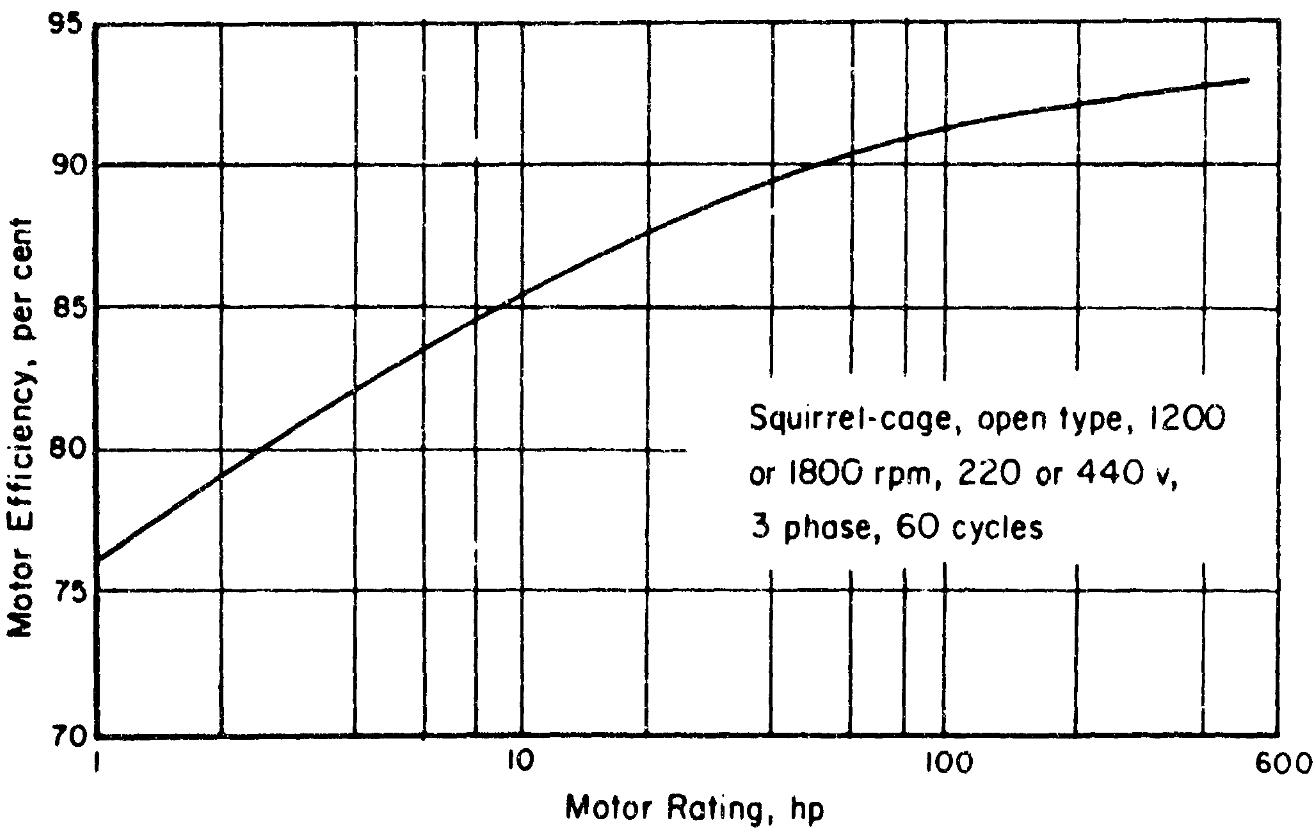


FIGURE 51. ELECTRIC MOTOR EFFICIENCY

Tab'e 11 lists the insulating materials found to be most satisfactory in resisting damage or deterioration from moisture.[15] These materials also would be useful in other components of the auxiliary power system for moisture resistance.

TABLE 11. ELECTRICAL INSULATIONS FOR HUMID CONDITIONS

Material	Application
Alumina	Ceramics
Asphalt	Varnish
Dacron-glass	Textiles (with silicone)
Epoxy	Enamel, lqiuid resin, varnish
Formvar	Enamel
Glass	Textiles (with silicone)
Irrathene	Plastic tape
Mica block	Ceramics
Mica paper	Papers
Paint	Filled resins and varnishes
Phenolic (modified)	Varnish
Polyester	Liquid resin
Polyethylene	Films, plastic tape
Polyurethane	Enamel
Porcelain	Ceramics
Silicone	Varnish, resin
Silicone rubber	Plastic tape
Steatite	Ceramics
Teflon	Films
Thermoplastic	Molded plastics
Urethane	Liquid resin

Note: These materials are also capable of withstanding short-time immersion in water.

Maintenance requirements of the generators can be reduced further by using brushless, rotating, rectifier-type equipment. Such equipment is used exclusively on railroad refrigeration plants and for similar applications where minimum maintenance and highest reliability are essential.[16] However, the high temperature limit for the semiconductor components used in this type of equipment is 150 F.[17]

If some moisture is absorbed between exercise periods, but not a sufficient amount to produce failure at rated voltage, the equipment can be operated for a period of time sufficient to ensure that all the absorbed moisture has been driven out. According to one manufacturer, it may take as long as 4 hours at full load to thoroughly eliminate absorbed moisture. If the other shelter equipment, including the prime mover, could be properly exercised in much less than 4 hours, the 4-hour drying period for the generator or motors would be a disadvantage.

There are several tests which will give indications of the relative condition of the electrical insulation in generators and motors and which will also suggest when equipment should be rewound, replaced, or simply dried out.[18] These tests are briefly described in References (18) through (23). Preliminary tests for moisture content of the insulation are important to prevent breakdown upon rapid application of full voltage. There is a basic weakness in all electrical tests because the usual causes of low-voltage equipment insulation failure are mechanical stresses and physical deterioration rather than electrical breakdown and all electrical tests are conducted on static equipment. However, in shelter a applications, physical deterioration and mechanical stresses should not be serious problems; however, moisture may be a serious problem.

Study of the various electrical tests reveals that the a.c. high-voltage test gives the best indication of the condition of the equipment. It is usually recommended that this test be preceded by low-voltage tests to reveal the extent of absorbed moisture that may have to be dried out before high potentials can be safely applied. However, a recently introduced voltage-breakdown tester[24] senses component breakdown, removes the voltage, and shorts the leads microseconds before damage can occur. This tester may, therefore, be used directly on the electrical equipment without preliminary low-voltage testing.

The reliability of electric-generator power-transmission systems can be expected to be high if proper inspection and operating procedures are observed. Most air-cooled generators, designed to National Electric Manufacturers Association (NEMA) specifications, will operate continuously at rated load with an ambient air temperature between 40 and 50 C (104 to 122 F). The temperature rise in the generator windings above ambient temperature is limited by the class of the insulation used in manufacture. If ambient temperature in excess of these limits must be tolerated, the generator can be either derated, operated with reduced life, or supplied with a higher-temperature insulation. In some cases a combination of these would be the best solution.

Following are values for the normal rated allowable temperature rise for various electrical insulation types. These data are based on 40 C ambient temperature and include a 10 C allowance for hot spots.

Class of Insulation	Rated Temperature Rise, C
A	50
B	80
F	105
H	125

For every 10-C increase in ambient-air temperatures, the allowable generator-winding temperature rise should be reduced approximately 10 C to maintain the same maximum insulation temperature. Since the maximum temperature rise for Class A insulation is 50 C, a 10-C reduction would be equivalent to about a 20 per cent reduction in the kilowatt output of the generator. Derating is obviously not practical for a very significant increase in ambient air temperature.

Generator-winding life is approximately 35,000 hours at rated load and 40 C ambient temperature conditions. Substantial increases in ambient air temperatures can be withstood if a much shorter life expectancy is permitted such as a

2-week operating period in a community shelter. As a rule of thumb, the insulation life is approximately halved for each 10-C increase in winding temperature. For instance, Class A insulation would have an average life of about 15,000 hours at 65 C (150 F), or an average life of about 1,500 hours at 100 C (212 F). It is suspected that shortening the operating life of the generator in this manner would have some effect on the reliability, but no specific information along this line is available in the literature or from manufacturers.

Figure 52 shows the effect of temperature on average life of generator and motor windings for three commonly used classes of insulation.[25] These data show that a higher temperature insulation would allow a substantial increase in ambient-air temperature and still possess long life. For example, Class F insulation has an average life of 35,000 hours at 100 C ambient temperature or 1,500 hours at 140 C (284 F) ambient temperature. The 100 and 140 C ambient-temperature limits should be modified in practice by a 10-C allowance for hot spots and a 5-C allowance for reduced heat-transfer rates at the elevated air temperatures. Thus, the ambient-air temperatures could be allowed to reach approximately 85 C (185 F) for a 35,000-hour life and approximately 125 C (257 F) for a 1,500-hour life with the generator operating at full rated load. Although generators could be operated at temperatures up to 125 C with reduced life, the maximum temperatures at which prime movers can operate satisfactorily is much lower than 125 C.

Electric generators are available for power outputs from a fraction of a kilowatt up to about 1,000 kilowatts in semiportable form. The size range availability varies from 1/2 to 1 kilowatt in under 10 kilowatt sizes to 5 to 50 kilowatts in the larger sizes. There is relatively little hazard in using electric-generator power-transmission systems as long as reasonable and proper precautions are taken during the installation. The noise level of such a system is very low.

Figures 53 and 54 show approximate purchase prices for electric generators and motors, respectively. As in the case of the efficiency curves of Figures 50 and 51, the generator cost curve is plotted against the electrical kilowatt output of the generator, and the motor cost curve is plotted against the shaft horsepower output of the motor. It is not possible to determine with any reasonable degree of accuracy the total cost of an electric-generator power-transmission system. or, for that matter, any power-transmission system, without knowing the specific details of the application.

Hydraulic Power-Transmission System

Figure 55 is a schematic illustration of a hydraulic power-transmission system. The basic system consists of a fluid pump driven by the prime mover, hydraulic lines to deliver the high-pressure fluid throughout the shelter, and fluid motors to convert the energy in the fluid to power for driving the mechanical equipment of the shelter. The pump speed is matched to that of the prime mover, usually between 1200 and 1800 rpm, for the convenience of direct connection. A clutch is necessary between the prime mover and the pump to allow starting under no load. A pump delivery pressure of around 1,000 psig would best match available industrial equipment. The basic types of available fluid pumps and motors are: gear, vane, and piston.

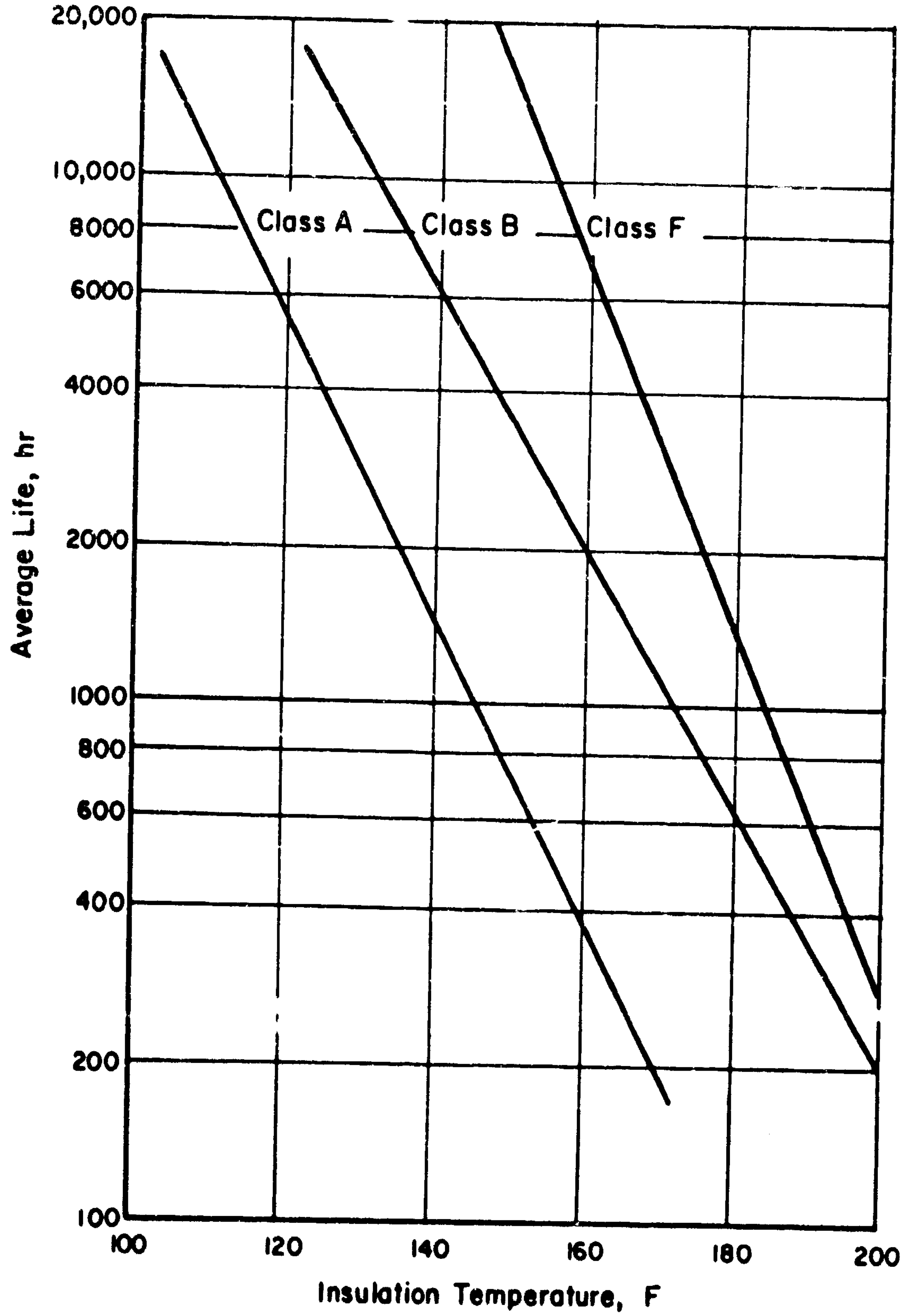


FIGURE 52. AVERAGE WINDING LIFE AS A FUNCTION OF INSULATION TEMPERATURE

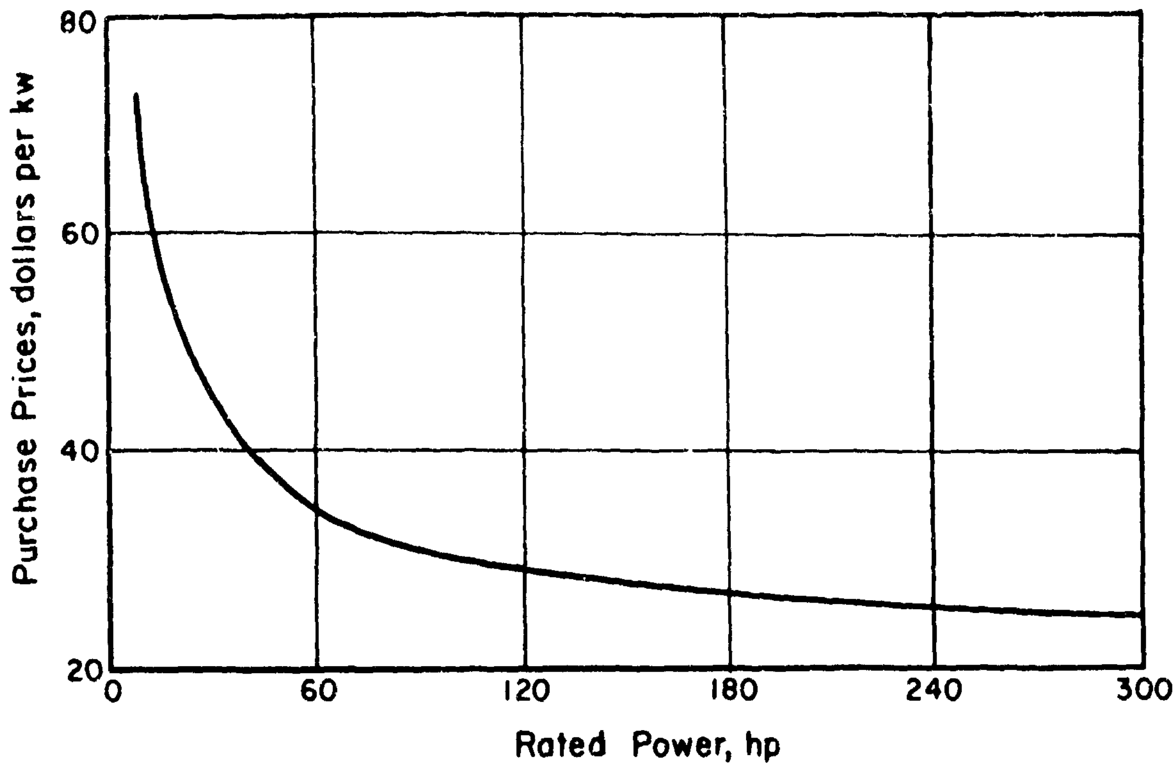


FIGURE 53. APPROXIMATE PURCHASE PRICES OF ELECTRIC GENERATORS

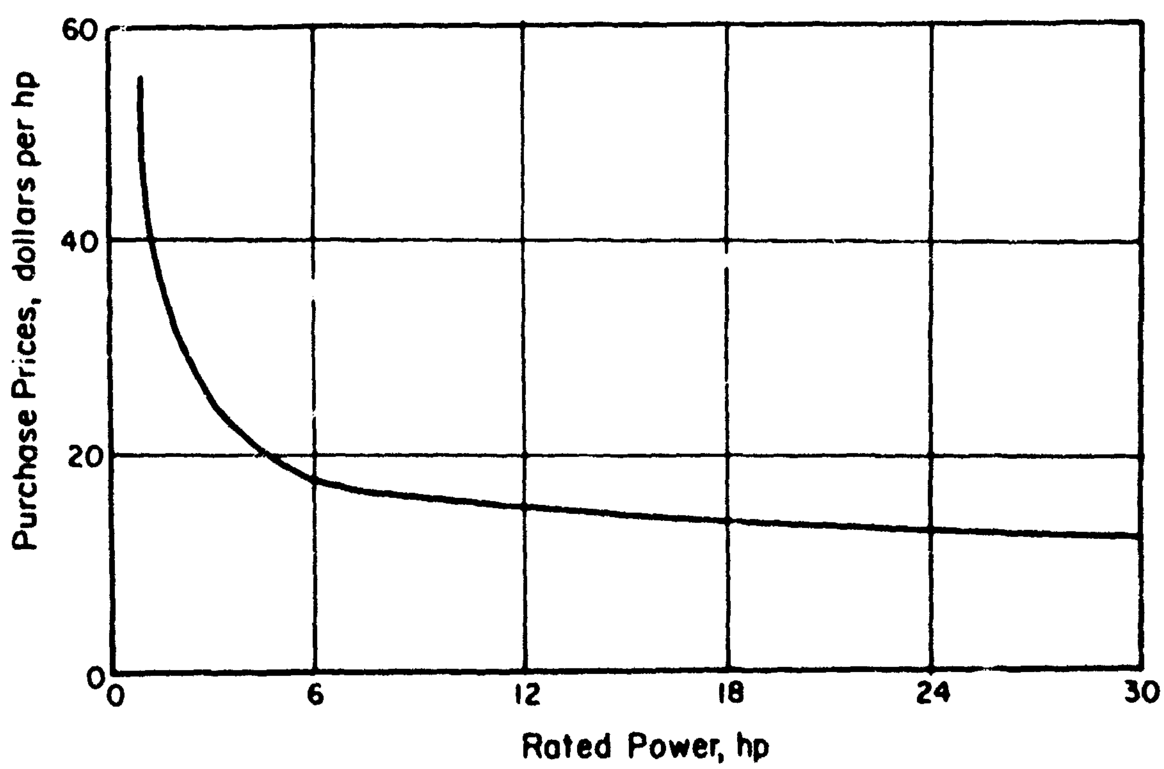


FIGURE 54. APPROXIMATE PURCHASE PRICES OF ELECTRIC MOTORS

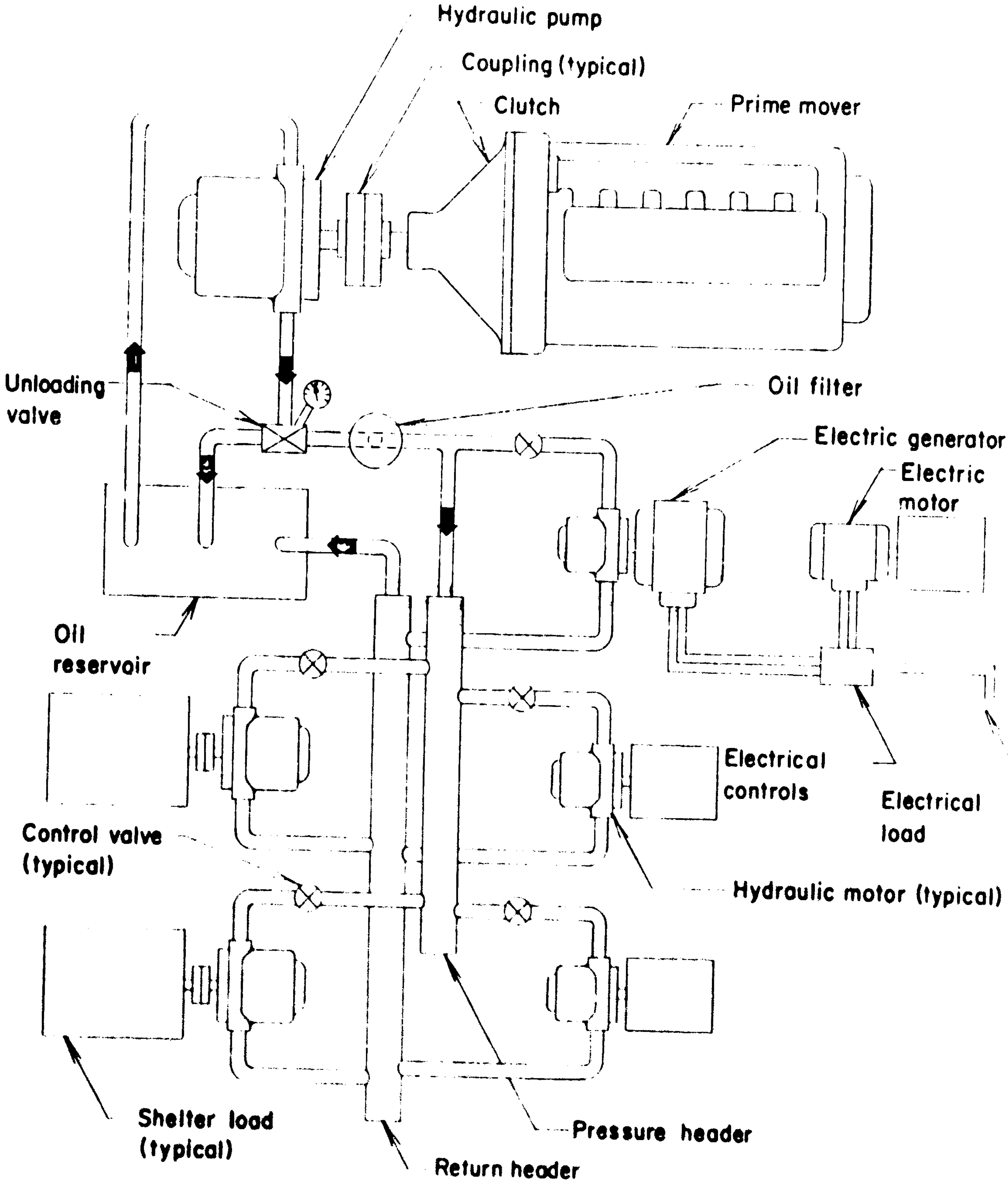


FIGURE 55. HYDRAULIC POWER-TRANSMISSION SYSTEM

The efficiency of hydraulic systems varies only slightly according to the size and somewhat more according to the type of components used. The efficiency range of gear pumps and motors is 60 to 90 per cent, of vane pumps and motors 75 to 90 per cent, and of piston pumps and motors 80 to 95 per cent.[26] Considering the pumps and motors by themselves, the over-all efficiency of a hydraulic power-transmission system would be fairly close to that of an electric-generator power-transmission system. However, losses in the transmission lines of a hydraulic system are significant. For example, if a 5-hp vane-type hydraulic motor is assumed to be located about 25 or 30 feet from the hydraulic pump and connected with 1/2-inch tubing, the total line pressure drop would be about 89 psi. The efficiency of power transmission through the lines would be 92 per cent. Using larger sized tubing would reduce the losses. A compromise would have to be reached between line efficiency, total fluid in the system, and total cost of the system.

Assuming a pump efficiency of 88 per cent, average motor efficiencies of 82 per cent, and line losses averaging 6 per cent, the over-all hydraulic power-transmission system efficiency would be approximately 66 per cent.

The primary maintenance requirement for a hydraulic power-transmission system is to monitor the condition of the filters and the contamination and oxidation levels of the hydraulic fluid. The filters should be changed or cleaned before the pressure drop across the elements becomes sufficient to open the by-pass valves. A common design pressure differential is 15 psi. Metal filter elements will remove particles down to about 10 microns in size, and active-earth filters will remove particles down to 3 to 5 microns. The active-earth filters also remove moisture and tend to neutralize acidity, but they cannot be used with some types of fluids because they also remove valuable additives. A sample of fluid should be taken periodically for examination to determine its condition. More frequent analysis would be required with petroleum-based or emulsion-type hydraulic fluids, because their storage life is shorter than that of the synthetics or water glycols.

Industrial hydraulic equipment is designed for full-load life expectancies of 2,000 to 5,000 hours for gear pumps and motors; 3,000 to 6,000 hours for piston pumps and motors.[26] These life expectancies are determined primarily by bearing life. Excessive bearing wear results in a reduction in pump output pressure. When cavitation is permitted, such as when there are excessive fluid velocities in the lines or high viscosities, the life of the equipment will be much shortened.

The operating temperature of the fluid at the pump inlet should be kept below a temperature at which the fluid viscosity becomes less than 70 SSU so as to maintain proper lubricating qualities. At startup, the fluid viscosity should not be greater than 4000 SSU to avoid cavitation. Therefore, the temperature limits of a hydraulic system will depend on the fluid chosen. Industrial equipment and fluids are normally designed to operate in ambient-air temperatures of 40 to 90 F.[27]

The reliability of a hydraulic power-transmission system could be expected to be very good if good quality components are used, if the system is well constructed, and if the proper hydraulic fluid is selected. In blast shelters, hydraulic piping must be flexibly mounted so that it will not be damaged by ground shock. Numerous additives are available to improve the hydraulic fluid, particularly in reference to reliability of the system. Among these additives are rust inhibitors, antioxidants, anticorrosives, foam inhibitors, and detergents. There are four basic types of hydraulic fluids which could be used in a hydraulic power-transmission system: water-glycol, synthetic, emulsion-type, and petroleum-based hydraulic fluids.

The water-glycol hydraulic fluids contain about 40 per cent water, the remainder being glycol synthetic thickener and additives for corrosion resistance and increased lubricity. No oxidation will occur if the pH factor remains about 7.5. The alkalinity factor indicates when the additives need replenishing to prevent oxidation. The viscosity of this type of fluid is between 150 and 300 SSU at 100 F, with a maximum temperature limit being 150 F and the recomended operating temperature being 120 F. The viscosity can be estimated by measuring the water content. Absorptive-type filters cannot be used because they absorb the additives. Because of the nature of the fluid, special paints, seals, and metals must be used throughout the system. Epoxy paints are best and seals made of rubber, buna N, butyl, neoprene, or leather can be used. Zinc and cadmium cannot be used and aluminum should be anodized. To compensate for the reduced lubricating qualities of water-glycol fluids, bearings must be oversized. For this reason, pumps of the balanced-vane or axial-piston types, which reduce bearing loads, are the most satisfactory.

Synthetic hydraulic fluids include phosphate esters and chlorinated hydrocarbons. These fluids do not oxidize or form sludge upon long contact with air, and they never break down during operation; thus, they do not require changing. The viscosity range available is 50 to 1,000 SSU at 100 F, with the most widely used being about 230 SSU. The normal maximum temperature limit is around 180 F and the recommended operating temperature is 130 F. These fluids are heavier than water and the line water traps must be inverted from the normal position for petroleum-based fluids. Water is quite easily separated as it does not mix with the synthetics. The lubricating quality of the phosphate esters compares very favorably with that of the mineral oils and, therefore, they are more commonly used than the water-glycol types. Special paints, pipe sealants, and seals will be essential with synthetic fluids. Paints which can be used include the epoxys, hard-cured phenolics, and nylons. Seals should be made of butyl, Viton A, or silicon. Aluminum should be anodized. Screened, expanded-metal, or active-earth types of filters can be used. Synthetics are designed for closed systems in particular and are compatible with all types of pumps when proper seal materials and paints are used.

Emulsion-type hydraulic fluids are water-in-oil emulsions and they contain about 40 per cent water. Lubricating qualities are fairly good because of the oil content, but not as good as those of the phosphate esters or mineral oils. The maximum temperature limit is 150 F if excessive water losses due to evaporation are to be avoided, and the recommended operating temperature is 120 F. The life of equipment in which emulsion-type fluids are used will be less than that of equipment in which regular oil is used because of the decreased lubricating qualities of the emulsion-type fluids. Particles smaller than 150 microns will remain in suspension in emulsion fluids. Because these fluids are oil-based, they will deteriorate with age. Most seal materials are compatible with emulsions, but special paints are required.

Petroleum-based hydraulic fluids are the least expensive of the four types discussed and are, therefore, the most commonly used when fire resistance and long-term storage with minimum maintenance are not important factors. These fluids break down and become less viscous with use. Oxidation, corrosion, foam, and rust inhibitors should be added. The maximum operating temperature is limited by the viscosity changes of the fluid and is related to the pump design, but may be around 200 F for some types. Viscosity-index improvers thicken the oil, and these types are more subject to breakdown of the polymer chains. The presence of viscosity-index improvers such as methylacrylate, which cause adhesion between particles, actually increase contamination-particle growth by their very nature of

agglomerating small particles.[28] These contaminants might clog system filters. Growth of particles can be minimized by initial filtering of particles that may later combine, by using polar rust preventive agents, and by not using methyl-acrylate viscosity-temperature improvers.

Hydraulic system components and fluids suitable for a community shelter power-transmission system are readily available. The primary hazards associated with hydraulic power-transmission systems are fire and fumes. The water-glycol and synthetic hydraulic fluids have the greatest fire resistance and the petroleum-based hydraulic fluids have the least.[29] Oil fumes might become objectionable in the shelter if an oil line or connection failed and hot oil came in contact with any hot surface.

Hydraulic machinery, particularly high-speed, high-pressure equipment, is somewhat noisy. The vane or gear pumps and motors are particularly noisy. However, the hydraulic motors driving mechanical equipment in the shelter would not be very large and, consequently, they could be enclosed in sound-absorbing housings. It is desirable, however, to keep the fluid velocities in the suction lines below 5 feet per second and below 15 feet per second in the supply lines.[26] The hydraulic motor speeds should be kept below about 2,000 rpm, not only to reduce noise but also to reduce shock forces, flow losses, and erosion in the system.

The costs of hydraulic pumps and motors vary appreciably but will be approximately as follows:[26] gear pumps and motors $3 per horsepower; vane pumps and motors $3.50 per horsepower; and piston pumps and motors $6 per horsepower. Hydraulic fluids also vary in cost depending on the type and on the additives. In general, the cost of the various hydraulic fluids will be approximately as follows:[30] synthetics, $3.50 per gallon; water-glycols, $2.25 per gallon; emulsions, $1.25 per gallon; and petroleums, $0.50 per gallon. The total cost of a hydraulic power-transmission system would be approximately equivalent to the total cost of an electric-generator power-transmission system. The cost of the basic components for the hydraulic system is considerably lower than the cost of the basic components for the electric-generator power-transmission system; however, the cost of the auxiliary equipment including hydraulic fluids, accumulators, reservoirs, and pipe-lines, and the cost of installation are both considerably higher for the hydraulic power-transmission system.

Some electrical power will be required in the shelter for lighting and communications; consequently, a small electric generator will be required. This auxiliary power generator could be driven by its own small prime mover, by the main prime mover also driving the hydraulic pump, or by a separate hydraulic motor as shown in Figure 55.

Pneumatic Power-Transmission System

Figure 56 is a schematic illustration of a pneumatic power-transmission system. The basic system consists of an air compressor directly driven by the power source, and air motors located throughout the shelter as needed for driving the various items of mechanical equipment. As in the case of the direct-drive and hydraulic power-transmission systems, an auxiliary electric generator will be necessary for lighting and communications. This auxiliary generator could be direct-driven from the main prime mover, be driven by a small prime mover of its own, or be driven by an air motor.

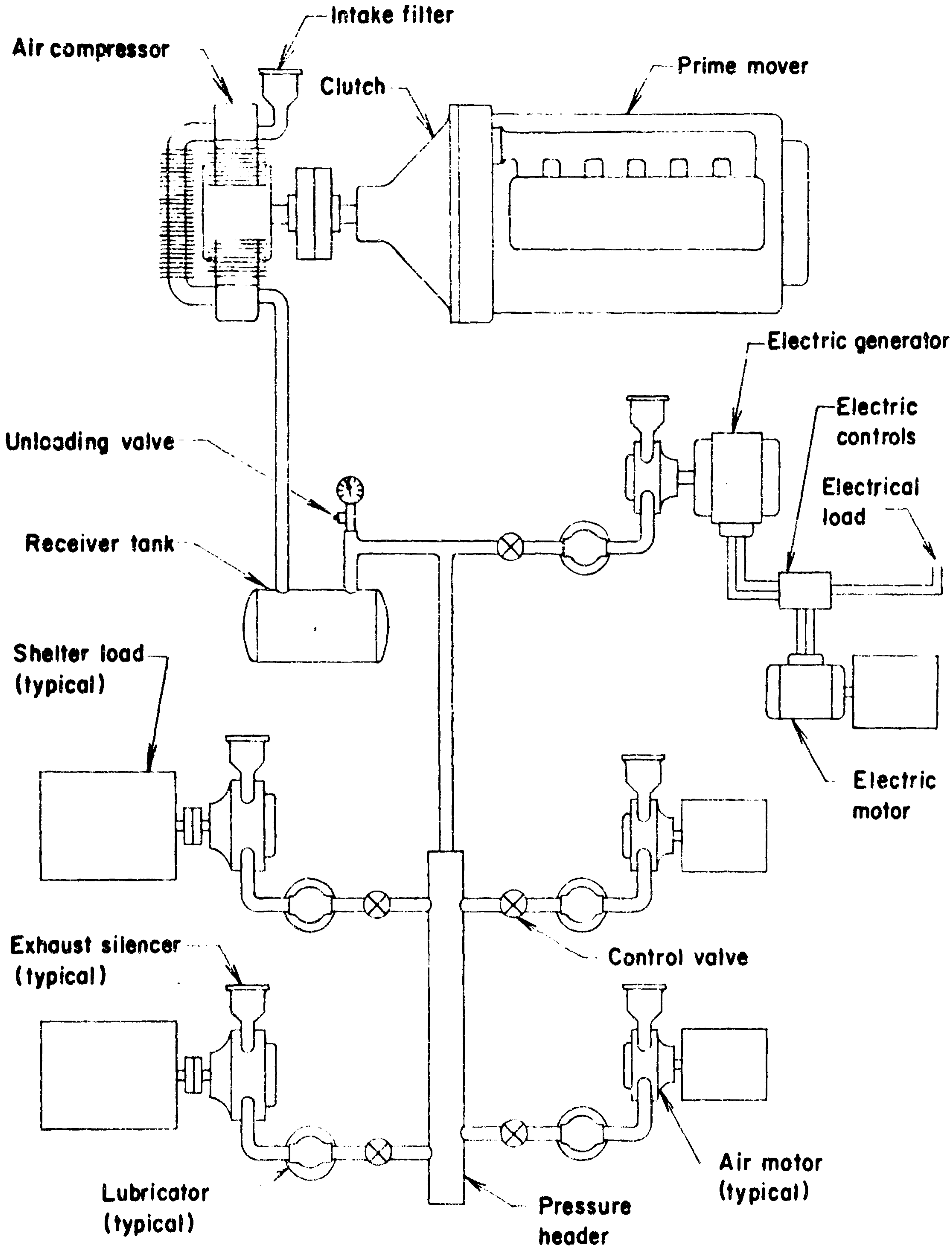


FIGURE 56. PNEUMATIC POWER-TRANSMISSION SYSTEM

The speed of the air compressor should be in the range of 1200 to 1800 rpm to match the operating speeds of most prime movers. A system pressure of 90 to 100 psig would best match commercially available air compressors and motors. A receiver tank should be used and the compressor should be designed oversize to permit a 50- to 80-per cent duty cycle to avoid overheating and to allow operation at high ambient air temperatures.(31) The time required to achieve maximum receiver-tank pressure for an air-cooled compressor should range between 10 and 30 minutes. The rest time, or unloaded condition, should be approximately the same. These times should be recorded at each exercise period because they are an indication of the fitness of the compressor and will reveal when repairs are needed. The 10-minute minimum is to prevent condensation buildup in the crankcase, and the 30-minute maximum is to prevent overheating which would shorten operating life and create an explosion hazard.

Since a great deal of heat is generated when air is compressed to 100 psig, a cooling means must be provided. Commercial air compressors are available, either air or water cooled. A water-cooled unit would be practically independent of ambient-air temperatures. If the prime mover itself is water cooled, its coolant could also be circulated through the air compressor and only one cooling system would be required. A water-cooled compressor would also be more efficient than an air-cooled unit because the compression process more nearly approaches isothermal conditions.

The efficiency of a well-designed air compressor is approximately 83 per cent. Commercially available air motors, on the other hand, are relatively inefficient mechanisms; for instance, a 5-hp air motor is only about 17 per cent efficient. Consequently, the over-all efficiency of a pneumatic power-transmission system, neglecting line losses which are likely to be small in a well-designed system, would be only about 14 per cent.

Air motor and compressor efficiencies increase very little with increasing equipment size and, therefore, no substantial improvements in efficiency can be realized by using large-size equipment, if, for other reasons, the use of several small units would be advantageous. The conventionally used air motor expansion ratio of 1.67 cannot be appreciably increased because the air entering such a motor at 90 psig and 100 F will be discharged at 54 psig and only 37 F. Further expansion of the air would cause ice crystals to form within a motor; therefore, the efficiencies of presently designed air motors cannot be improved upon for this application unless nearly all of the water vapor is removed from the compressed air. Dessicants such as silica gel can lower the dew point temperatures to -50 F or -100 F if necessary, and depending upon the quantity employed, can last 3 or more hours. Nevertheless, the standard commercially available equipment is nearly all limited to the 90-psig, 100 F design conditions which limit pneumatic system efficiencies to about 14 to 18 per cent over-all.

During stand-by the following items of pneumatic power-transmission systems should be checked periodically: oil levels in the compressor and air-line lubricators, safety valves, minimum-maximum pressure controls, intake air filters, and cleanliness of heat-exchanger surfaces. Idle periods of up to 1 year can be tolerated if preservative oil is distributed through the compressor and all openings are sealed.(31)

If exercising is planned, the recommended frequency is every two or three weeks. During exercising, the equipment should be run at full load for 30 minutes and then run unloaded for 15 minutes. Safety valves should be tested during

exercising to ensure reliable operation. Safety valves are generally set at 10 per cent above the maximum operating pressure of the system and should be so located that excessive line pressure buildup would not occur should any downstream valves be closed. A safety valve is frequently installed on the inter-cooler between stages of a multi-stage compressor to prevent damage to the inter-cooler due to leakage from the high-pressure stage.

Over-lubrication of the compressor cylinders during exercise and emergency running should be avoided because of the increased rate of deposit buildup on the valves, cylinder walls, and piston rings which increases maintenance time. Also, the higher concentration of oil vapor and the incandescence of the deposits increases the likelihood of explosions. Under-lubrication will cause rapid ring wear and over-heating due to increased friction.

From the standpoint of reliability, the pneumatic power-transmission system is satisfactory for community shelter application. Air motors should have a life expectancy of approximately 2,500 to 3,000 hours between overhauls. A piston compressor should last approximately 5,000 hours without maintenance.

The pneumatic system components which would be required for a community shelter are fairly readily available. There should not be any serious safety hazards associated with pneumatic power-transmission systems. Fire-resistant crank-case oils could be used to reduce the possibility of creating an explosion or fire hazard in case of an oil leak. However, these fire-resistant fluids require special seal materials, increased lubrication rates, removal of paint from inside surfaces, and sometimes inversion of water traps because of a specific gravity greater than that of water. Rust and oxidation inhibitors should also be used in the crankcase oil because some moisture is bound to condense in the crankcase, especially if the equipment operates in a humid environment.

Air systems are generally quite noisy, especially air motors, and this would be particularly undesirable if the discharge were into the shelter space. Air will be exhausted at approximately 54 psig; therefore, there will be sonic velocity at the exhaust port. A silencer on the outlet would be almost essential. The exhaust air will contain oil which may become objectionable in the shelter over a two-week period. An oil trap incorporated in the silencer may control the concentration of oil adequately. Otherwise, the exhaust air will have to be piped outside the shelter or back to the engine room.

As was mentioned previously, the air entering a conventional air motor at a temperature of 100 F and a pressure of 90 psig would leave at a temperature of 37 F and a pressure of 54 psig. Allowing this air to expand to atmospheric pressure would further reduce the temperature to about 33 F. This air could then be used for shelter cooling. Only 4 F of additional cooling is obtained in the free expansion because no work is being done by the air. By calculation it can be shown that one hundred cfm of this cool air (the approximate air consumption of a 3-hp motor) entering the shelter at 33 F and leaving at 85 F would provide about 5700 Btu/hr of cooling. This cooling is obtained only if the air leaving the compressor is cooled prior to distribution to individual air motors. If no cooling is done at the compressor the net effect of the pneumatic power-transmission system is to raise the shelter temperature as occurs with the electric and hydraulic power-transmission systems.

Pneumatic equipment costs about the same as electrical equipment on a per-horsepower-output basis. However, the pneumatic power-transmission system over-all efficiency is only about 1/5 that of an electric-generator power-transmission system. Consequently, the size of the prime mover, the quantity of fuel to be stored, and the capacity of the heat sink would have to be increased by a factor of five.

SYSTEM MOUNTINGS AND DRIVES

The selection of the system mountings and drives will have a significant influence on the performance and reliability of the total power system installation. Differences in engine weights, vibration characteristics, component alignment requirements, and driven loads greatly change the mounting and drive requirements. There are various techniques available, each having advantages and disadvantages in specific applications. Practically all of the techniques would be applicable in some degree to community shelter auxiliary power systems, but a few would be more appropriate than the others; hence, a review of the basic considerations and pertinent equipment is presented in this section.

General Considerations

There are four main items to be considered in designing or selecting mounting and drive methods for an auxiliary power system: (1) alignment, (2) vibration, (3) service connections, and (4) load type.

For direct coupling, the alignment between the driving and the driven components (prime-mover and power-transmission system) must be maintained within reasonably close tolerances. Misalignment can cause bearing or structural failure in either component, and excessive vibration. Out-of-balance vibration would be undesirable when transmitted to engine room floor and walls, especially so when it is passed on at an annoying level to the shelter occupied space. Too much vibration of the prime-mover and power-transmission system components may cause component or instrument failures. Torsional vibration, caused by the torque pulses of a piston engine, the inertia of the driven load, and the elasticity of the shafts and connecting member, could also be destructive to the components of the system.

Service connections such as fuel lines, water lines, and exhaust and intake ducting should be sufficiently flexible in the vicinity of the prime mover that relative movement between prime mover and floor or wall will not cause fatigue failure in these parts. The type of load will have some bearing on whether torsional vibration will be a problem, and it will also indicate whether a clutch will be needed.

Mounting

The means of mounting prime movers and driven components fall into two general categories: foundation mounting and skid mounting. A foundation mounting is generally a large mass of concrete or other heavy material upon which the prime mover and driven component are independently mounted. With skid mounting, on the other hand, both components are mounted and aligned on a common steel-rail framework.

A mounting is considered flexible if the prime mover or the prime-mover and driven component combination is free to move. Movement, or vibration, must generally be allowed in both horizontal and vertical planes. A prime-mover foundation can be flexibly mounted by isolating the concrete mass from the floor and ground with resilient material such as cork or sand and gravel. Alternatively, spring or rubber elements can be interposed between the prime mover and the

foundation to provide freedom of movement for the prime mover alone. This latter type of mounting arrangement imposes a severe duty on the coupling between the prime mover and the driven component whether the driven component is flexibly mounted itself or not.

Skid mounting is preferred from the standpoint of the coupling requirements as the alignment between the prime mover and driven component is not affected whether the entire unit is rigidly or flexibly mounted.

A mounting is considered rigid if no measurable freedom of movement for the prime mover is allowed for. This type of mounting is satisfactory for small piston engines and generally for all sizes of gas turbines. Most driven components, such as generators and rotary pumps or compressors, may also be rigidly mounted. Reciprocating pumps and compressors and larger piston engines should not be rigidly mounted because of the undesirability of the transmitted vibrations and because of the shock forces within the equipment itself.

In general, skid mounting would be preferable for all community-shelter auxiliary-power systems, with the skid flexibly mounted on the floor of the engine room. A built-up foundation or strengthening of the floor section upon which the unit is mounted would not be necessary for any but the very largest of piston-engine installations that might be considered for community shelters.

Drive Methods

Drive methods pertinent to community shelters are: direct coupling, indirect coupling, and clutch. A clutch may be used with either direct or indirect coupling.

Direct coupling may be a solid connection between the two shafts, requiring extremely accurate alignment or it may be a flexible connection which permits a small amount of misalignment. The solid connection is not recommended because of the cost involved and the skill required for installation. Flexible couplings are available in many different forms. Among the design points to be considered when a flexible coupling is selected are: shaft end play, angular and parallel misalignment, shaft speed and torque throughput, and prime-mover and driven-component vibration characteristics.

Indirect coupling means include belts, chains, and gears. These devices require less critical alignment of the shafts and generally provide adequate flexibility to accommodate out-of-balance and torsional vibration. Gears and chains may be noisy unless fully enclosed. The use of belts, chains, or gears imposes side loads on the shafts and bearings of the prime mover and driven component. The equipment must be designed for the extra loads or else outboard bearings must be provided to share the load and stiffen the shaft assemblies.

A belt, chain, or gear drive is frequently selected because the shaft speeds of the prime mover and driven component do not match. Indirect coupling, particularly a belt or chain drive, frequently makes integral skid mounting of the prime mover and driven component difficult.

For most community-shelter auxiliary-power-system applications, the direct-coupling method will be the most satisfactory. In many instances, the driven component, particularly if it is a generator, can be designed to share a common housing bolted directly to the prime mover so that shaft alignment and rigidity are not problems.

A clutch will be necessary between the prime mover and driven component if the load requires high torque upon startup or at low speeds, because the starting ability of prime movers is limited either to no load or to a load which builds up slowly as speed increases. In addition, a clutch is desirable if it is necessary to rotate the prime mover shaft or driven component shaft independently for servicing or for warm-up. Centrifugal pumps and compressors and electric generators can be coupled directly to the prime mover for starting. Piston and rotary-vane pumps and compressors have high starting torques and, hence, require clutches so that they can be disconnected from the prime mover for starting.

NOISE AND VIBRATION

Noise for the purposes of this study is considered to be any undesirable sound. Noise generated by shelter auxiliary power equipment, if uncontrolled, would be undesirable mainly because of psychological effects on the shelter occupants. The noise, even though it might be quite loud, would not damage hearing; however, because it would be continuous and would have an unchanging frequency spectrum, it would be extremely annoying to most individuals and probably unbearable to some.

Vibration of the shelter structure, on the other hand, will not be a major problem. Vibrating equipment can easily be isolated from the structure and, in addition, the rigidity of the structure would tend to damp out vibrations.

The prime mover in any auxiliary power system will be a major source of noise. Piston engines produce "broad-band" noise. Gas turbines tend to produce more nearly pure tones at high frequencies, i.e., they "whine". Depending on the type used, the power transmission system may also be a major source of noise. A piston air compressor used in a pneumatic system would be about as noisy as a piston engine prime mover. A hydraulic system pump would be relatively quiet as would an electric generator. Fortunately, the use of more than one noisy component does not greatly increase the noise level above that associated with the operation of the noisest component. For example, if two components having noise levels of 65 and 68 dB were used together, the resulting noise level would be 69.8 dB.

Noise control can best be achieved in the initial design of the system by placing the noise-generating equipment in a separate, tightly sealed enclosure located as remotely from the occupied space as possible. With the proper selection of mounting methods, equipment vibrations will not be transmitted through the shelter.

Sound levels are measured with respect to the source and also with respect to the receiver. The source is usually a vibrating object, but it may also be an aerodynamic pulsation. At any given instant, the source will be radiating a certain amount of sound power. This sound power level (PWL) is a function of the source and is not changed significantly by the environment of the source. Sounds measured with respect to the receiver are expressed as sound pressure levels (SPL), usually in units of dynes per sq cm. The sound pressure level is greatly influenced by the size and absorptivity of the environment of the source and also by the distance of the receiver from the source.

The human ear is not equally sensitive to sound pressure at all frequencies and for this reason a useful description of a source must specify frequency as well as sound pressure level. For instance, to the average listener a 70 dB tone at 4,000 cps sounds as loud as an 85 dB tone at 100 cps.

Although it will be necessary to reduce the noise level associated with the operation of auxiliary power systems, it is desirable to have a moderate noise level in the occupied space in the shelter for masking purposes.

The study of shelter noise and vibration problems was divided into three parts: (1) determination of noise criteria, (2) evaluation of methods of noise control, and (3) experimental investigation of vibration and noise reduction. The experimental work was carried out in the Battelle underground test facility

with the 20-kw demonstration unit. The tests and results are described in the Demonstration Unit section of this report.

Noise Criteria

Because judgments concerning the undesirability of any sound are almost completely subjective, the process of determining noise criteria necessarily involves averaging the responses of a large number of persons. The statistical nature of noise criteria precludes prediction of whether or not one individual will be annoyed by a certain noise, but allows prediction of the percentage of people that will be annoyed. The relative objectionability and speech interference produced by noise is expressed in the form of noise criteria curves. These curves approximately express the response of the human ear to pure tones of various frequencies.

Figure 57 shows noise criteria (NCA) curves[32] for applications where costs are of primary importance or when the people present have some important reason to be tolerant of the noise produced (as is the case with an auxiliary power system providing lighting and ventilation). These curves represent a compromise between loudness and the cost of control measures for noise which is steady and free of beats between low-frequency pure-tone components. From these curves and from published noise criteria data for rooms[32], it would appear reasonable to specify an upper limit of NCA 55 as a noise rating for a shelter.

Methods of Noise and Vibration Control

A noise problem can be divided into three aspects: the source, the path, and the receiver. The source, for the purposes of this discussion, is an auxiliary power system with vibrating surfaces. The path by which the noise travels from the source to the receiver is either directly through the air or through solid objects to some radiating surface, and then through the air to the receiver. The receiver is the shelter occupant.

A noise problem can sometimes be prevented or reduced by proper selection of equipment. The designer should consider noise output as one of the factors in evaluating components for use in an auxiliary power system, even though more important considerations may dictate final selection of a noisy unit. Standard procedures for measuring sound power in standard frequency bands have been established but little or no information is presently available on the sound power level of commercially available prime movers. It is possible, however, that if the sale of a large quantity of engines were at stake, the manufacturers would obtain and release noise information.

Treatment of the noise path is the major possibility for shelter designers in their efforts to control noise. The most common method of noise reduction is placing a barrier or wall between the noise source and the receiver. Two qualities are of major concern in a wall used for sound reduction: massiveness and good sealing against pressure leaks. A wall which does not meet the ceiling will do more good than no wall at all. However, sealing up the last small opening can increase effectiveness to a far greater extend than the percentage of

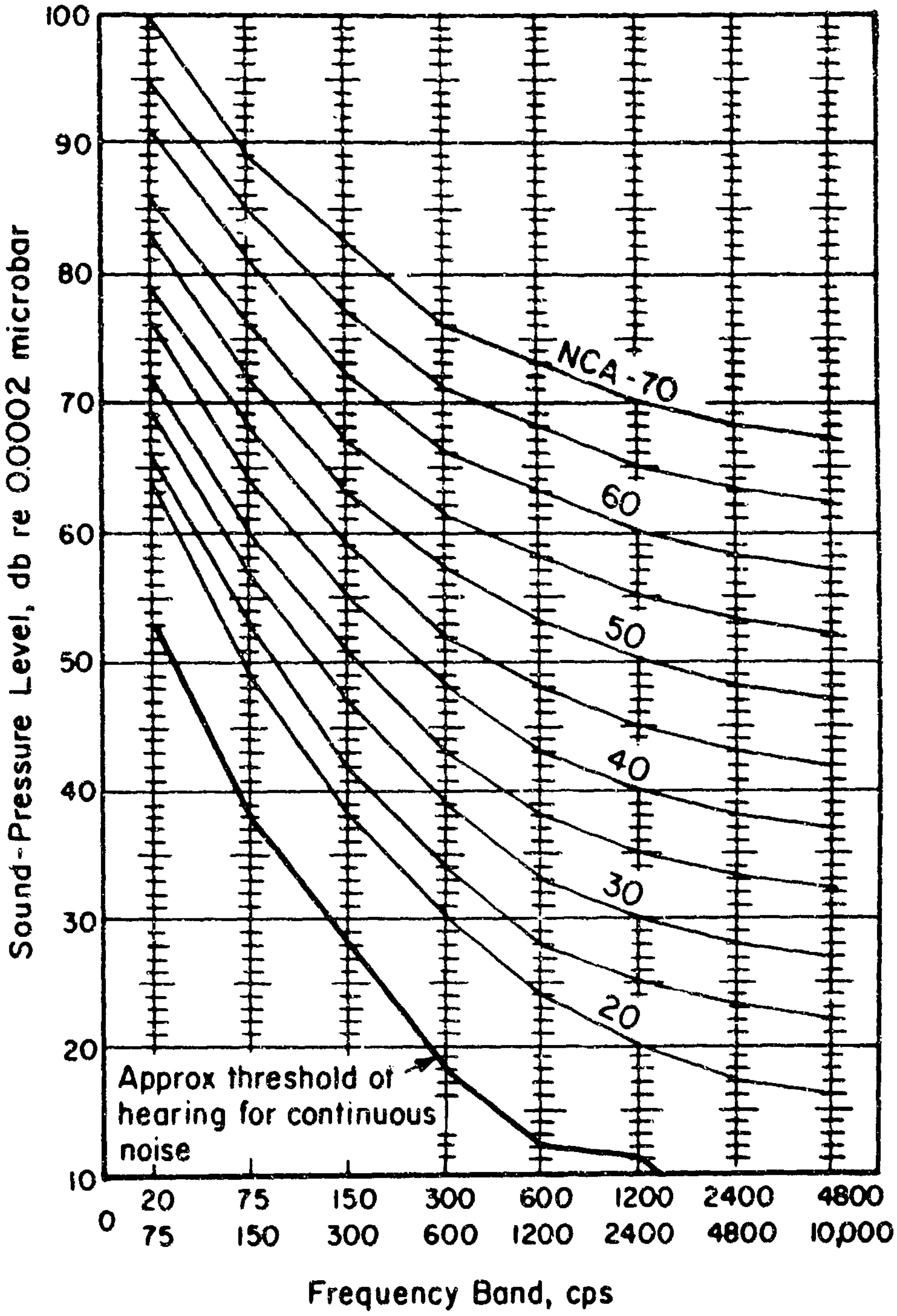


FIGURE 57. ALTERNATE NOISE CRITERIA (NCA) CURVES

wall involved might seem to indicate. It is not possible to achieve a noise transmission reduction of more than 20 dB from one room to the next if the dividing wall has leaks amounting to 1 per cent of its area common to both rooms. To reduce sound leaks substantially, it is necessary to caulk around pipes passing through walls and to use door gaskets that seal on all four edges

of doors. Poor design practices, such as installing electrical boxes back to back in two adjacent rooms, should be avoided.

In general, any crack that will leak air will also leak sound. An effective sealing does not have to withstand high pressures, for 120 dB SPL amounts to only about 0.003 psi. Porosity will also permit leaking of sound, particularly those of higher frequency. Cinder-block walls present particular difficulties in this respect. However, a coat of plaster or heavy paint will eliminate such leaks.

Mass is the one characteristic which will always reduce the sound transmissivity of a pressure-tight wall. Increasing the rigidity may also reduce transmissivity at some vibration frequencies, but it will increase it at others. Other techniques, such as applying vibration-damping material to the wall, or building a light-weight wall close to but isolated from the vibration of the original wall, can sometimes be used to advantage. Fortunately, the concrete or masonry walls one would expect to find in shelters have sufficiently low transmissivity for good noise reduction.

Noise can also be reduced effectively by using two walls separated by an air space. Such an arrangement is effective because each transfer of acoustic energy from the air to the wall and then from the wall to the air on the other side is inefficient due to the mismatching of acoustic impedances of air and a solid wall. The transmissivity is low, but transfer occurs only twice for a single wall no matter how massive or tightly sealed it is. With two walls, transmissivity is halved, but only if the walls are sufficiently far apart. If they are close to each other, the acoustic coupling between them becomes more effective. A nominal separation distance is 2 feet. This naturally suggests the simple expedient of planning storage areas or other unoccupied rooms between shelter engine rooms and living areas.

If it is necessary to have an opening for air flow between the engine room and the living area, a "sound trap" duct should be utilized. Sound traps are commercially available and are essentially long, narrow passages lined with acoustical absorbing materials to attenuate the sound passing through them. A typical passage might be 7 feet long with an open area 4 by 30 inches. Such a sound trap has an average attenuation of over 40 dB in all frequency bands. Simply lining a metal duct with glass fiber bats will provide a good degree of attenuation, but commercial sound traps are optimized for maximum attenuation and minimum air-flow losses for a given volume. Performance descriptions are available from manufacturers.

The air around the receiver (listener) is the final portion of the noise path. Because much of the noise reaching the listener has been reflected from walls and objects in the room with the listener, these surfaces are considered as a part of the noise path. The difference in sound pressure level which can be obtained by changing from minimum to maximum sound absorption of these surfaces is only 6 dB. Consequently, the noise must be controlled primarily elsewhere in the path. However, most common acoustical materials such as ceiling tile, upholstered furniture, and various cloth objects have maximum sound absorption in the range of 500 to 4,000 cps. This is also the frequency range in which the human ear is most sensitive. Hence the absorptive materials tend to reshape the noise spectrum (loudness level versus frequency) to one less objectionable.

Finally, noise may also be reduced through vibration isolation. In shelters of concrete or masonry the direct transmission of equipment vibration through the structure to radiating surfaces in the living area should not be a problem. Simple isolation procedures (such as the use of commercially available rubber pads under vibrating equipment) will reduce vibration transmission. However, in shelters constructed of materials other than concrete or masonry, the vibration isolation problems will be more complex and require more sophisticated isolation procedures.

The possibility of vibration transmission through pipes or other paths which connect engine room and living areas should be considered by the designer and avoided. For example, if engine cooling water is used to heat water in the living area, it is likely that vibrations would travel through the water pipes. This could cause noise by exciting resonant frequencies of the heat exchanger or water tank. A solution would be to install flexible vibration isolators in the interconnecting piping close to the engine.

Treatment of the noise receiver is not so straightforward. About the only common technique in this instance is to provide acoustic "masking" in the form of steady, broad-band, background noise. This has been used to afford speech privacy and to cover annoying sound from typewriters and other intermittent devices in situations where noise control is inadequate because of light-weight wall partitions, etc. A certain amount of acoustic "masking" is provided in most buildings by ventilating and air-conditioning systems.

In a shelter, the somewhat muted noise from the engine and/or circulating fans is likely to be the main source of acoustic "masking" to keep people from being annoyed by other people's conversation, babies crying, etc., and to give them some degree of privacy for their own conversation. For this reason, some low level of equipment noise (perhaps NCA 30 - 35) in the shelter area would be desirable.

STAND-BY MAINTENANCE

The auxiliary power system of a community shelter would be much like any emergency stand-by power system in that it probably would not be called upon to serve its function for many years. Yet, it must be ready to perform reliably at all times. Reliability in a somewhat unusual sense is required. Instead of having long service life, the equipment must be able to tolerate long idle life without loss of ability to start and operate.

Requirements for Stand-By Maintenance

Equipment stored for long periods of time must be protected against structure deterioration causing structural weakness and against surface fouling causing inoperability. Structure deterioration may result from corrosion, rust, rotting, and mold. All of these are accelerated to a considerable degree in the presence of moisture and oxygen. Surface fouling is caused by gum and sludge formation, sedimentation, and deposition of solids where fuels, lubricants, and other liquids are stored within the equipment or are used as preservatives. Airborne dirt may also cause surface fouling. In general, a number of measures are desirable in any stand-by maintenance program: control of the storage atmosphere, cleanliness of the parts, and high purity and deterioration resistance of the liquids, lubricants, and preservatives used in the system.

In addition, it is necessary to guard against the loss of stored liquids and gases through leakage and evaporation. In some cases sufficient reserve liquids or gases can be provided to make up normal losses. In other cases, such as in hydraulic or pneumatic starting systems, normal leakage over a long period of time might reduce the reliability of the system substantially.

Additional problems which may arise from long-term storage include loss of storage-battery charge in the engine starting system and fretting corrosion in bearings and other close-fitting parts which remain in one position for a long time and which are subject during that time to vibration.

Maintenance Techniques

There are two general approaches to stand-by maintenance: dynamic maintenance and static maintenance. Conventional emergency stand-by power systems are almost universally dynamically maintained. Dynamic maintenance involves frequent and periodic inspection and exercising. Static maintenance, on the other hand, involves preservation of the equipment during long-term storage with, perhaps, fairly frequent inspection and only infrequent exercising.

Most conventional emergency stand-by power systems are expected to start immediately and to deliver full power in just a few minutes when an emergency occurs. There is great need, consequently, for a state of readiness. Starting must not be delayed by even the simplest preparatory steps such as removing the covers and seals and pumping of fuels, lubricants, and coolants into their

respective systems. Many stand-by systems start automatically when commercial power fails and assume the full load within a matter of seconds. The criticality of instant starting in installations of this type no doubt has exerted a strong influence on the established pattern of stand-by maintenance. Another important factor in most conventional installations is that experienced personnel are readily available for stand-by maintenance. Hospitals, communications facilities, hotels, apartment buildings, and many other similar installations have building superintendents or maintenance men who are capable of maintaining and operating emergency stand-by power equipment. Thus, the cost of a dynamic maintenance program is easily absorbed in the expense of general maintenance.

On the other hand, the auxiliary power system in a community shelter probably would not require immediate starting in an emergency. Moreover, maintaining a staff of trained personnel may be economically impractical. A delay of up to 3 or 4 hours in placing the auxiliary power system into operation would usually be tolerable if long-shelf-life batteries or a small, easily started, portable engine-generator set were available to supply the basic lighting and communications requirements of the shelter. Also, a community shelter system might comprise many relatively small and widely dispersed neighborhood shelters. This could present a formidable physical and economic problem in a dynamic maintenance program. For these reasons, serious consideration was given during this investigation to the possibilities of long-term preservation of the equipment and infrequent exercising and inspection--that is, to a static maintenance program.

The results of studies on static maintenance by personnel of the U.S. Navy and the Canadian Army are presented in this section of the report.

Dynamic Maintenance

Dynamic maintenance, as previously stated, is employed almost universally for conventional emergency stand-by power systems. The chief reasons for frequent exercising and inspection are to evaporate accumulated moisture from the system, to distribute lubricants and fuels over critical surfaces, and to check for actual or potential trouble spots.

Manufacturers and users of emergency stand-by power systems do not agree on a single most appropriate frequency of exercising. Frequencies varying between once a week to once every 6 weeks are common. The most desirable frequency for a given installation would naturally depend upon the specific equipment in the system, the temperature and humidity conditions prevailing in the system enclosure, and the extent of any precautions taken to reduce the amount of moisture and oxygen coming into contact with the equipment.

The components and subsystems of the auxiliary power system should be inspected at least as frequently as the system is exercised. The equipment should be inspected for: (1) evidence of leaks at valves, fittings, gaskets, etc., (2) unusual noises in operating mechanisms, and (3) soundness of fastenings, couplings, covers, etc. Samples of the fuel, lubricants, coolants, and other liquids in the system should be inspected for appearance in comparison with standards or with previous samples. Items to look for in these samples include color or opaqueness and suspended solids or sediment. The electric generator and electric motors should be given a voltage check as described in the section of this report dealing with power transmission systems.

A significant ingredient of a successful dynamic maintenance program is the recording of data during the exercising of the system. The engine performance data of most significant value are speed, load, and temperature of the coolant, the lubricant, and the exhaust gases.

The power output of the engine during exercising must be inferred indirectly from the performance of the rest of the power system. For instance, the power output of an electric generator could quite easily be measured with inexpensive instruments, and then an average generator efficiency could be assumed to determine the power output from the engine. The engine power output with pneumatic or hydraulic power transmission systems would not be so easily measured; however, a reasonably useful measure of the power might be obtained with simple flow- and pressure-measuring equipment.

It is not as important to obtain the absolute power output as it is to obtain a record of any changes during the life of the installation. If the engine speed and fuel-rack or throttle positions do not change from one exercise period to the next under equivalent load conditions, the engine power capability can be assumed to be adequate.

Tests that should be made regularly but at less frequent intervals than exercising include: (1) testing of the fuel and lubricants for sulfur and gum content, corrosiveness, and residue after distillation, and (2) testing of the coolant for corrosiveness.

Static Maintenance

Static maintenance has been practiced largely in situations where the time and effort involved in preparation for storage and in subsequent reactivation were not restricted. Warehouse storage of goods is one example; packaging of machinery and equipment for overseas shipment is another. Generally, an artificial atmosphere conductive to low rates of deterioration is maintained, as in the warehouse, or the individual equipment items are thoroughly coated with a preservative material, as for overseas shipping.

The end of World War II triggered a very rapid and thorough demobilization of our armed forces. To guard against being caught unprepared once again, and to salvage some value from the tremendous expenditures that went into war material which never saw service, considerable work was done to devise improved means for preserving the more costly items of military equipment for possible future use with minimum reactivation effort. Out of this work came the term "mothballing", which means long-term preservation.

In 1946, the U.S. Navy sponsored a laboratory test program[33] in which samples of materials and equipment normally found on board ships were stored under controlled humidity conditions for a period of about 4 years. Duplicate sample groups were stored in sealed cabinets with 15, 30, 45, 65, or 90 per cent humidity conditions and with no temperature control. Following the storage period, the items were inspected and in many cases tested for strength and performance.

Among the conclusions drawn from this test are: (1) all material should be clean before storage to minimize corrosion and mildew, (2) a 30 to 35 per cent humidity level is the best compromise for most materials, (3) critical metal

surfaces and parts should be coated with a preservative for added protection, and (4) limited corrosion does not appear to reduce the strength of materials.

In mothballing ships, the Navy has followed two approaches. For some ships entire decks or holds are sealed off from the atmosphere and a dehumidifying system is installed to maintain a predetermined humidity condition throughout the space. For other ships, the mechanical equipment is individually sealed in cocoons and dry circulating air is supplied to the cocoons from a central dehumidifying system. All equipment and surfaces not included in the sealed cocoons are coated with a preservative which is periodically renewed.

Reactivation after mothballing is usually an involved process, requiring 30 days or more, depending on the size of the ship or the equipment. During the reactivation process, parts or components may be replaced, if inspection indicates the need, before the equipment is put into operation. Though semiskilled or unskilled personnel are frequently employed in the reactivation job, a knowledgeable person is usually in charge.

The Navy experience discussed above is not directly applicable to the community shelter program because of the involved procedures and extensive equipment. However, certain minimum requirements for long-term storage were demonstrated and the mothballing techniques which are developed, though they do not provide for rapid enough reactivation, may be considered a reasonable starting point for developing community shelter preservation techniques.

A recent technical paper reported on some experience in Canada involving outdoor storage and reactivation of military vehicles. This experience apparently applies a little more directly to community shelter problems and requirements(34). The authors describe an experiment conducted with a fleet of Canadian army trucks which had been stored outdoors for 6 to 8 years. The purpose was to evaluate the effectiveness of the preservative techniques being used at that time and to ascertain how well the vehicles would perform with a minimum of reactivation effort.

The trucks were stored on blocks, the tires deflated to one-third normal pressure, and the cabs sealed. The engines were run briefly before storage, the fuel being unleaded gasoline. Run-out was on preservative engine oil, which was also subsequently sprayed through the spark plug holes into each cylinder. Preservative oil was used in the crankcase; operational lubricants were used elsewhere (transmissions, differentials, etc.). Every 3 years the engine and gear lubricants were changed, and once each year during the last 2 years of storage, the power trains were exercised by manual rotation. Reactivation involved primarily inspecting and checking brakes, lights, windshield wipers, and other accessories. The only actual work done on the vehicles was that which was considered essential for safety.

A dozen vehicles, six 3/4-ton trucks and six 2-1/2-ton trucks, were reactivated in the spring and were driven, fully loaded, over a course consisting mostly of pavement, but also including 80 miles of gravel per 1000 miles, for a total of about 10,000 miles during the entire summer. Another four vehicles, two 3/4-ton trucks and two 2-1/2-ton trucks, were reactivated in the winter and driven in a manner to simulate stop-and-go operation for a total of about 3,800 miles. Each of the 16 vehicles was then driven at approximately 15 miles per hour for 200 miles over cross-country terrain.

All of the vehicles tested were adequately serviceable upon reactivation. The only serious deficiency noted during the tests was the seizure of brakes on two of the vehicles. The cause was found to be reduction in the boiling points of the brake fluids as a result of an accumulation of moisture in the brake systems.

The main conclusions drawn from this program are: the preservation techniques and materials used were effective in retarding deterioration in storage and the reactivated vehicles were adequately serviceable.

The basic requirements for a community-shelter static-maintenance program, based on the Navy and Canadian experience, would appear to be: (1) clean all components and surfaces before preparation for storage, (2) coat critical surfaces with preservative fluid, and (3) maintain a relative humidity of 30 to 35 per cent during storage.

Some types of auxiliary power-system components may integrate better into a static-maintenance system than others. For instance, an LPG-fueled gas turbine, utilizing its fuel in the gaseous state, would present few or no problems because of gum and varnish deposits from fuel left in the fuel system during storage. There would also be few problems because of bearing, cylinder wall, and piston failure due to corrosion, etc., from the atmosphere or from lubricating or preservative oils or because of rust, or corrosion in the cooling system. Since LPG fuel has the best storability characteristics of the commercial fuels, it would be a logical choice for a static-maintenance program. A manual energy-storage starting system would require no attention or special maintenance equipment and, therefore, it should be well suited to a static-maintenance program.

The chief value of a static-maintenance program for the auxiliary power system of a community shelter would be a cost saving during the potentially long, idle stand-by period. With less actual handling and operation of the equipment the possibility of human error affecting the reliability of the system would also be reduced.

Scheduling Maintenance

During this program, time was not sufficient for a thorough investigation of static maintenance. However, to establish some measure of the cost, complexity, and effectiveness of a static maintenance program as compared with a dynamic maintenance program, the available information on long-term preservation was used with appropriate qualifications to formulate static maintenance procedures for a sample community-shelter auxiliary power system. The suggested dynamic maintenance procedures presented in this report were formulated for the same sample community-shelter auxiliary power system.

The following specifications were arbitrarily chosen for the sample shelter:

Occupancy	- 750 persons
Space	- 15 sq ft per person
Prime mover	- 100-hp turbocharged diesel engine
Power transmission system	- Electric generator and motors
Fuel	- Kerosine in sealed, underground tank
Cooling system	- Ebullient with steam vented to atmosphere
Starting system	- Hydraulic.

Dynamic Maintenance Schedule

The basic elements of dynamic maintenance are frequent inspection and frequent exercising. A suggested dynamic maintenance schedule for the sample community-shelter auxiliary power system is as follows:

(1) Exercise entire system for a total of 3 hours once every month (see typical exercise schedule for piston-type internal combustion engine below)

(2) Test generator and motors every 3 months with static-voltage test

(3) Sample and test fuel after 4 years and then at 1-year intervals

(4) Replace lubricants every year

(5) Replace fuel when tests indicate deterioration.

A typical exercise schedule for a piston-type internal combustion engine might be as follows:

(1) Start the engine under no load and warm it up with the load gradually being increased from light to full as the oil temperature increases to about 150 F

(2) Operate the engine at full load and rated speed with the oil temperature in excess of 210 F for from 30 minutes to 2 hours

(3) Remove the load except for the power required to operate the engine auxiliary systems with the engine still running at full speed

(4) Shut the engine down when the oil temperature drops below 140 F.

Allowing the engine to warm up under light and increasing load promotes faster warmup and thus reduces startup wear. Full-load operation of the entire auxiliary power system should be for a long enough period to assure that all of the moisture is driven out of generator and motor windings and all lubricating and hydraulic oil sumps. The length of time this will require will vary with the size and type of the system. Allowing the engine to cool down slowly is desirable to promote temperature equalization which reduces the danger of failure due to thermal distortion.

Exercising the auxiliary power system at full load may require auxiliary equipment such as a dummy electrical load or transfer switches to connect the generator output into the commercial power lines. The latter scheme would have to be arranged in cooperation with the local power company. On the other hand, it may be desirable to exercise all of the shelter mechanical equipment along with the auxiliary power system. For this purpose a dummy heat source would be required to load the air conditioning system. Steam from the engine ebullient cooling system piped through a simple heat exchanger inserted in the shelter ventilation system would probably provide sufficient heat for about 3/4 of the cooling load, which would be acceptable.

The actual cost of a dynamic maintenance schedule as outlined above cannot be accurately estimated without more intimate knowledge of many factors, most of which are likely to vary considerably from community to community. However, for comparative purposes a "rough" estimate for the first 5-year period is as follows:

Labor	$1,500
Fuel-exercising, testing, and replacement	630
Lubricants and other fluids	70
	$2,200

Static Maintenance Schedule

The basic element of static maintenance is control of the storage environment. A suggested static maintenance schedule for the sample community shelter auxiliary power system is as follows:

(1) Drain and flush cooling and lubrication system with light preservative oil

(2) Coat all critical surfaces with light preservative oil

(3) Store fresh lubricating oil in sealed container

(4) Seal engine room or entire shelter and maintain relative humidity at 30 to 35 per cent

(5) Inspect and exercise every 12 months

(6) Sample and test fuel after 4 years and then at 1-year intervals

(7) Replace fuel when tests indicate deterioration.

Preparation of the equipment and components of the auxiliary power system for stand-by storage might be as follows:

(1) Run engine for 5 to 10 minutes with preservative oil in crankcase

(2) Use preservative oil to purge the fuel system and spray some directly into the cylinders after the engine is stopped

(3) Drain the cooling system and flush with light preservative oil.

A potentially simple way to maintain the relative humidity in the shelter between 30 and 35 per cent would be to utilize a mechanical refrigerator-type package dehumidifier in conjunction with the air-circulation system in the shelter. During stand-by the static maintenance system should be capable of providing the proper relative humidity conditions regardless of outside conditions. Consequently, for this example schedule, a maximum outside air temperature of 96 F was assumed along with a relative humidity of 45 per cent. A total air change once a day was assumed to be adequate to maintain the proper conditions.

Calculations indicated that a make-up-air blower of 70-cfm capacity would be required, that a circulating air blower of about 140-cfm capacity would be adequate, and that a dehumidifier of about 10,000-Btu-per-hour capacity would satisfactorily remove the moisture from the air under the assumed conditions. The equipment involved in the stand-by maintenance system would be small compared with the equipment required for an emergency period. For instance, with the shelter fully occupied the make-up-air blower would circulate about 2250 cfm, the circulating air blower would circulate about 10,000 cfm, and the air-conditioning system would remove about 375,000 Btu per hour of heat.

It is quite possible that the make-up-air and circulating-air blowers could be the same for both stand-by and emergency periods if they were provided with two drive motors each: a small drive motor for the stand-by period and a large drive motor for an emergency period. The dehumidifying system would be separate from the primary air-conditioning system and would be similar to an ordinary home-type 1-ton window air conditioner.

The yearly schedule of inspection and exercising of the systems under the suggested static maintenance schedule would be similar to the monthly dynamic maintenance schedule. The entire system would be exercised, the generator and motors would be tested with the static-voltage test, and fuel and lubricant samples would be tested.

Reactivation from stand-by for exercising or actual emergency service would require merely that the cooling water be let into the cooling system, the preservative oil be drained out of the engine crankcase and the regular lubricant into it, the engine fuel system be primed with a hand pump, and the auxiliary power system be started.

The cost for static maintenance for the first 5-year period is roughly estimated to be:

Electric power	$400
Labor	200
Fuel-exercising, testing, and replacement	450
Lubricants and other fluids	50
	$1.100

In addition to the above recurring costs there would be an initial cost for air circulating and dehumidifying equipment. For the sample shelter this cost is estimated to be $300.

A comparison of the estimated costs for static and dynamic maintenance shows that static maintenance is potentially a much less costly method for maintaining auxiliary power systems in a state of readiness. Because of the many variables involved, the estimated costs shown for each maintenance method may not be indicative of the actual costs but it is felt that they accurately portray the costs on a relative basis.

DEMONSTRATION UNIT

The study of minimum requirements for auxiliary power systems would have been incomplete without experimental testing in a simulated shelter environment to verify the general specifications for design, installation, and operation which have been discussed in other sections of this report. Consequently, a diesel engine-generator set of 20-kw capacity was acquired and an experimental program was conducted. Information was gained from this experimental program which was not available in the literature. More severe operating conditions were imposed on the equipment than is recommended practice for the usual long-life installations. The main emphasis during the experimental testing program was on the effectiveness of various engine-cooling techniques and on minimum room ventilation rates required to maintain satisfactory operating temperatures.

Prior to the experimental testing program, visits were made to inspect several local installations of auxiliary power generators and to obtain information about operational experience. One visit was also made to a local group fallout shelter facility which contains a 100-kw diesel-electric power-plant installation. The unit is cooled with an engine-mounted radiator, and ventilation air for both the engine room and the cooling system is unfiltered. With this system, the engine room can become radioactively contaminated, but personnel are not expected to be required to service the engine during an emergency period. Provisions are made to add oil remotely and to automatically maintain the oil level in the crankcase. The jacket water is maintained at 120 F by an immersion heater in the engine block during stand-by. The engine-generator skid is isolated from the floor by vibration mounts; and the engine room walls are treated with sound-absorbing material to reduce the noise level in the occupied portion of the shelter.

Description of Equipment

The power unit selected for the experimental investigation was an Allis-Chalmers diesel engine-generator set. It incorporated a 35-bhp (continuous rating at 1800 rpm) naturally aspirated diesel engine with an Onan 20- to 35-kw brushless generator with Class B insulation. The unit was purchased with the following auxiliary equipment: electric starting system, voltmeter, ammeter, frequency meter, hour meter, high-water-temperature cutout, lo[illegible] pressure cutout, over-speed cutout, water-temperature gage, oil-pressure gage, muffler, 180 F thermostat, block-immersion heater, oil and fuel filters, and hand priming pump. An engine-driven d.c. generator provided power for charging the starting batteries and for the emergency cutout switches. The total cost of this equipment was $3,690 or $185/kw for 20-kw operation.

The generator output was connected for 120/208-volt, 3-phase, 4-wire, 60-cycle use and wired to a variable-resistance load bank. The Class B insulation used in the 20- to 35-kw Onan generator was sufficient for continuous 35-kw operation with a temperature rise of 80 C (176 F) at a normal ambient air temperature of 40 C (104 F). The temperature rise was only 50 C (122 F) at 20-kw output and 40 C (104 F) ambient; hence, substantial increases in ambient air temperatures could be tolerated without damage to the generator.

Figure 58 gives an over-all view of the experimental unit. The engine and generator are rigidly mounted to a skid and the controls are fastened to the generator housing and back panel plate. For the experimental tests, the skid was mounted on four rubber vibration-isolation pads, each having an area of approximately 15 square inches. Since the unit weighed about 2,000 lb, the pad pressure distribution was about 35 psi.

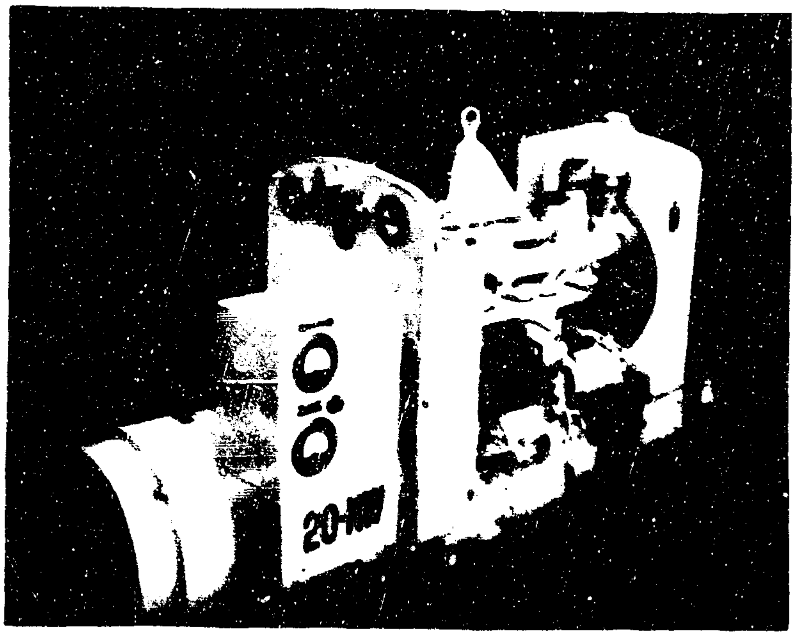
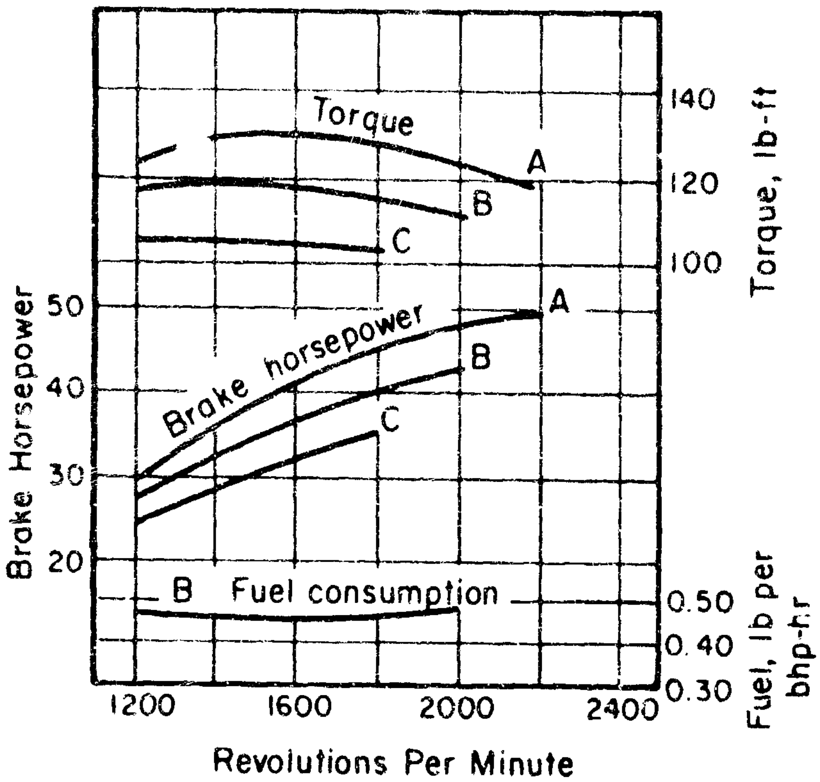


FIGURE 58. DEMONSTRATION UNIT, 20-KW ENGINE-GENERATOR SET

Figure 59 gives the published performance curves and the technical specifications for the diesel engine. The curves are corrected for sea-level conditions of 60 F ambient air temperature and 29.92 inches Hg barometric pressure.

The unit was supplied with the same radiator normally used for the four-cycle, 226-cu in. gasoline engine ordinarily used to drive the generator at its higher rated output of 35 kw. Therefore, the radiator was oversized for the 175-cu in. diesel engine which developed only 20 kw from the same generator. This extra capacity was advantageous during operation with minimum engine room ventilation.

Figure 60 shows the Battelle underground test facility used for the simulated shelter environment which was originally constructed for rocket research. It has a floor space 8 ft 8 in. by 6 ft 4 in. and walls 7 ft high. All sides are of poured concrete construction 8 in. thick, and the walls and ceiling are lined with 3-in.-thick oak planks. A 22-ft entrance hallway adjoins the room and is separated from it by a door and sound-absorbing duct through which ventilation air is passed. The entire structure, with the exception of the opening into the hallway from outside, is completely covered with a mound of earth to a minimum depth of 3 ft.



Engine Power Ratings

A MAXIMUM LABORATORY PERFORMANCE: The maximum power the engine will develop for five minutes without loss in speed. Production engines will deliver maximum power within 5 per cent of values shown.

B INTERMITTENT DUTY: This represents power available for applications having varying loads and speeds with full power being required for short periods.

C CONTINUOUS DUTY: This represents power available for driving sustained full loads for 24 hours per day operation.

ALTITUDE AND TEMPERATURE RATINGS: For approximate ratings, deduct from the intermittent or continuous rating 3 per cent per 1000 feet above 1000 feet altitude and 1 per cent per 10 degrees above 60 degrees F.

Engine Specifications

Bore and stroke	3-9/16 x 4-3/8 inches
Number of cylinders	4
Piston displacement	175 cubic inches
Horsepower (maximum)	49
Crankcase oil capacity	5 quarts
Radiator and block capacity	16 quarts
Net weight (including fan)	650 pounds
Compression ratio	15.35:1

FIGURE 59. DIESEL ENGINE CHARACTERISTICS
ALLIS-CHALMERS MODEL D-157

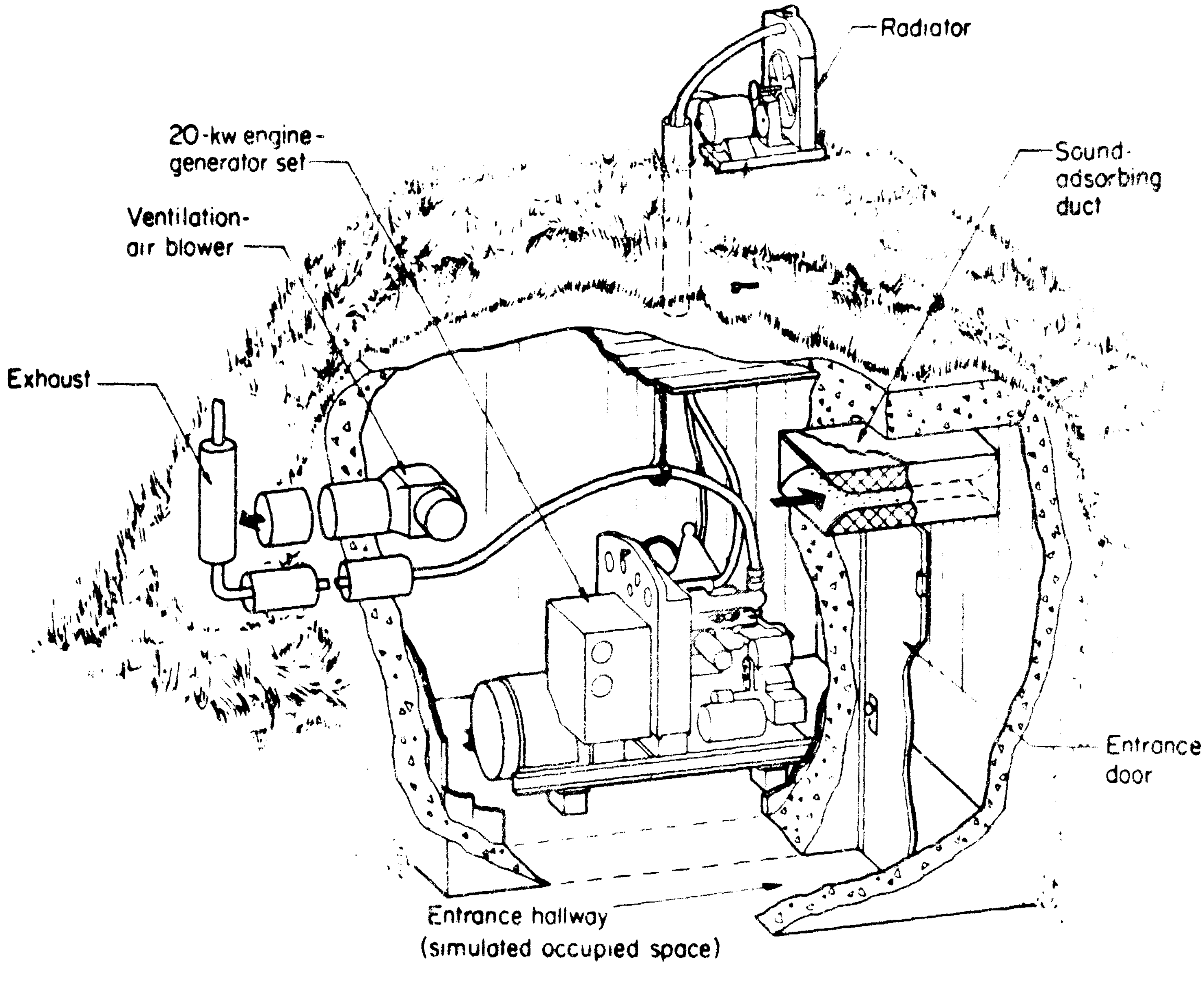


FIGURE 60. UNDERGROUND TEST FACILITY

Description of Instrumentation

Figure 61 shows some of the instrumentation and accessories used to monitor and record data from the performance tests. Additional parameters measured, which are not indicated on the figure, include oil-sump temperature, ventilation-air inlet and discharge temperatures, ventilation-air flow rate, generator insulation temperature, generator load, fuel rate, jacket water flow rate, and room wall temperatures. The temperature measurements were obtained using Chromel-Alumel, 1/8-in.-diameter, metal-sheathed thermocouples which were held in place by compression fittings. The temperatures were recorded on a Leeds and Northrup 12-point recorder and were also visually observed on a Brown 48-point indicator.

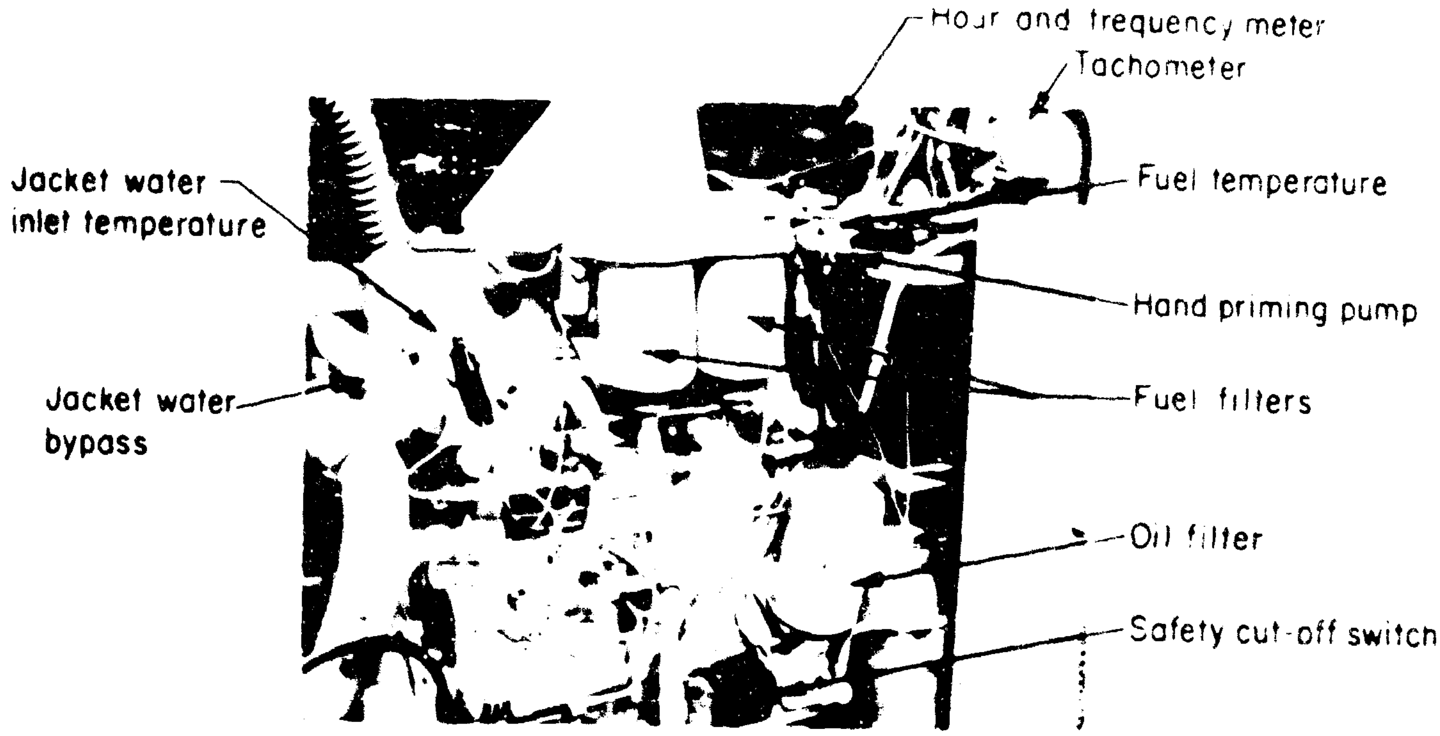
Two additional cutoff controls were added to the setup for greater safety. A manually operated shutoff switch was installed in the hallway and wired in parallel with the other safety cutout switches for closing the fuel solenoid valve when actuated. A second added safety control, also wired to the fuel solenoid valve, was a thermostat to sense the temperature of the air leaving the generator cooling system. This safety cutout was added to prevent the room air temperature from reaching excessive levels should the ventilation-air blower stop functioning during long, unattended test runs. Its location near the generator was chosen to assure a continuous air flow past the sensing element. The cutout temperature was set at 190 F. Since the generator cooling air temperature rose 15 F above ambient at 20-kw operation, the engine would be automatically stopped if the engine room ambient air temperature exceeded approximately 175 F. This safety control was not necessary when the radiator was engine mounted within the room because then the high-water-temperature cutout would stop the engine if the room air temperature reached excessive levels. However, this safety control was useful when the cooling system was independent of engine room air temperature, such as would be the case with an outside radiator, a water-to-water heat exchanger, or an ebullient cooling system.

Experimental Operation

The tests conducted during this experimental program fall generally into the following categories:

(1) Basic engine performance
(2) Ventilation system effects
(3) Exhaust system effects
(4) Maximum power performance
(5) Noise and vibration
(6) Exhaust waste-heat recovery
(7) Cooling system performance.

The basic engine performance tests were run with the engine-generator set installed as received to determine the significant operating parameters and their relationship to the manufacturer's ratings and specifications. During the ventilating system tests the quantity of the engine room ventilating air was varied to determine the effects of ventilation on engine performance. The engine



LEFT SIDE

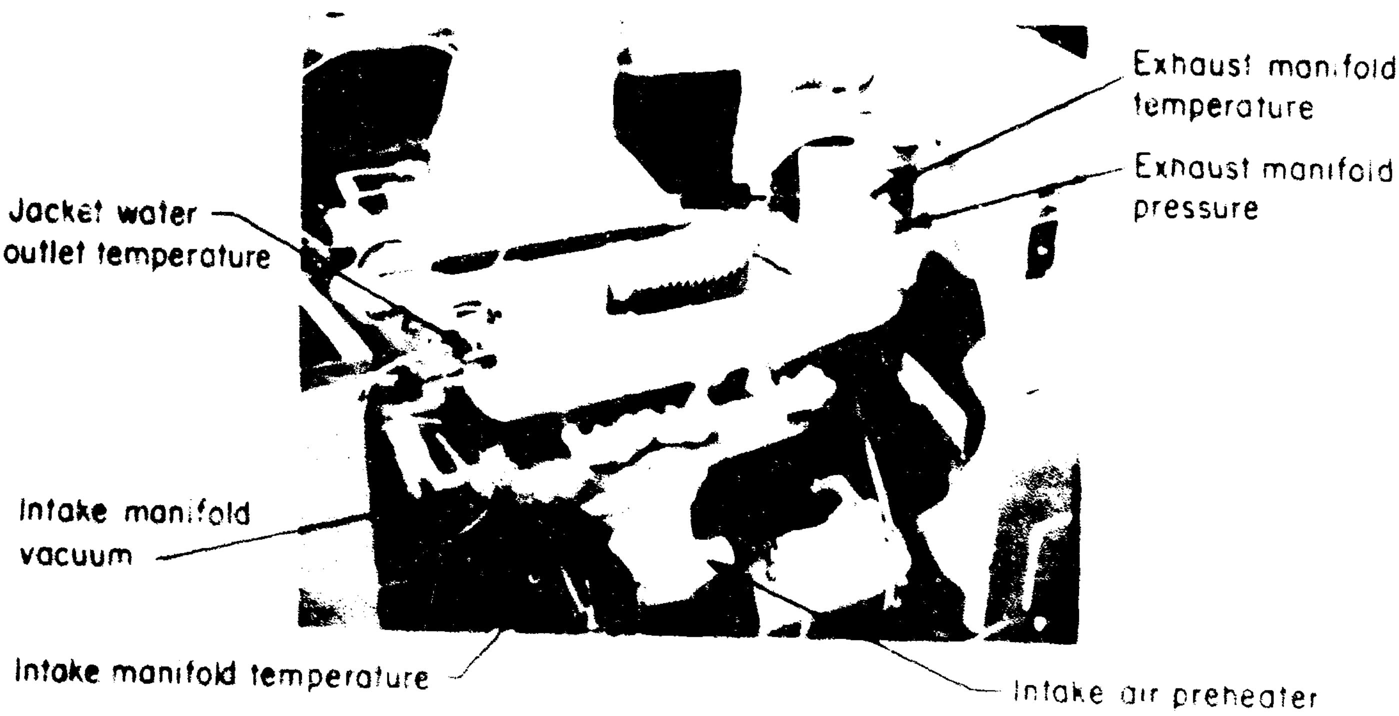


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FIGURE 61. DEMONSTRATION UNIT INSTRUMENTATION

room ventilation air was circulated either by the engine-driven fan or by independently driven ventilating blowers. The amount of ventilation air circulated was controlled by dampers.

Two long-duration test runs were made to establish equilibrium conditions within the test facility. These tests were conducted to obtain an average heat balance for the operating engine and to obtain a measure of the heat loss through the walls of the simulated shelter. The maximum-power performance tests were made by adjusting the governor externally for full output and increasing the load until the engine could no longer maintain 1800 rpm shaft speed. These tests were conducted to determine the effect of overloading on engine operating temperatures.

The noise and vibration tests were made to verify the adequacy of relatively simple control techniques. The exhaust waste-heat-recovery tests were made in conjunction with the waste-heat-recovery study. The conduct and results of these tests are discussed in detail in the waste-heat-recovery section of the report.

The cooling systems evaluated during the experimental test program are: an engine-mounted radiator, an outside-mounted radiator, a water-to-water heat exchanger, and a vented ebullient cooling system. The same radiator was used for both internal and external locations. When engine mounted, the radiator was cooled with a six-bladed pusher fan driven by the engine at 1.3 times the engine speed. The normal unrestricted air flow rate for this system was 3600 cfm at rated engine speed. When externally located, the radiator was cooled with the same fan driven by a 2-hp electric motor at 2600 rpm. The air flow rate for this system was estimated to be approximately 3800 cfm. The outside radiator location was 12 ft above its previous location on the engine skid; although the radiator was raised 12 ft, the engine water pump produced sufficient flow through the radiator so that no auxiliary water pump was needed.

The water-to-water heat exchanger was a shell-and-tube-type American Standard, Model 7M301-2D-6, and it was properly sized for the diesel engine used. At full generator load output of 20 kw the water flow rate through the heat exchanger, using city water at 60 F, was about 2 gpm. The cost of the heat exchanger was $150.

The ebullient cooling system consisted of a 20-gal tank used as a reservoir and steam separator. The tank was located above ground. The steam was vented to the atmosphere, and make-up water was supplied to the tank through a float-controlled valve. At full load the make-up water flow rate, using city water at 60 F, was about 0.2 gpm. The discharge line from the engine entered the steam separator above the water level to allow a more rapid steam separation.

The engine fuel used during most of these tests was No. 2 diesel oil; however, an alternative fuel, kerosene, was tested in the engine to determine its effect on the maximum performance capabilities of the engine.

The tests were conducted using fairly conventional procedures. Approximately 15 to 30 minutes was found to be necessary for stabilization of operation for each test condition. The actual time was dependent upon the nature of the test and the previous operating condition. After stabilization, data were recorded at 3- to 5-minute intervals for periods up to 30 minutes, again depending

on the nature of the test. Because of the great amount of data obtained during each test, it was possible to assess the reliability of the test and justifiable to ignore inconsistent data points if they did not occur often.

The vibration of the unit caused two failures during the test program, one of which may have been serious had it occurred in a shelter under emergency conditions. One failure was of a part modified by the local distributor and the other involved a generator control component which was used as supplied by the manufacturer. In the first instance, an oil fitting vibrated loose and allowed oil to drain from the engine. This failure emphasized the need for a low-oil-pressure cutout provision, even though on the demonstration unit the failure was detected before the engine was automatically stopped. The second failure was a broken wire in the generator control box caused by fatigue due to vibration after only 50 hr of running. The failure, which caused the loss of one phase of the three-phase output, could have been prevented by more adequate support of the wires in the control box. In an emergency it is doubtful that repairs could have been made.

A starting problem occurred which was very simple but which, under the stress of an emergency, might cause considerable difficulty. On several occasions attempts were made to start the engine without resetting the fuel cutoff switch because it was located out of view on the lower left side of the engine. A better location for this switch would be on the instrument panel adjacent to the starting switch. The need for "foolproof" starting procedures, controls, and interlocks cannot be overemphasized.

These failure experiences could be indicative of the need for a "shake-down" run of 50 to 100 hr immediately after installation to uncover potential failures due to faulty manufacturing, assembly, or installation work. Also it is recommended that all of the auxiliary equipment that does not have to be mounted on the engine-generator set be mounted separately to reduce the danger of vibration-induced failures.

Experimental Results

The results of the engine-generator set performance tests generally substantiated the manufacturers' claims and confirmed the performance predictions obtained from the literature. Essentially the same amount of heat was rejected from the engine to the cooling system regardless of the cooling technique used. Engine room ventilation had a significant influence on engine performance when the engine-mounted radiator cooling system was used.

At equilibrium operating conditions about 13 per cent of the total heat rejected from the engine-generator set was transferred through the walls and ceiling of the enclosure and to the surrounding earth. On No. 2 diesel oil the engine developed a maximum power output equivalent to the manufacturer's intermittent duty rating. On kerosene the maximum power output was about 3 per cent lower.

The information developed during the experimental program indicates that conventional prime movers can be expected to perform satisfactorily in a shelter environment if conventional installation practices are followed. The

results also show that full-load engine performance may be difficult to achieve with an engine-mounted radiator because of the large quantity of engine room ventilating air required.

Basic Engine Performance

Figure 62 shows the fuel consumption for the engine-generator set as a function of load. The scatter in the data reflects the influence of variations in combustion-air inlet temperatures, jacket-water and lubricating-oil temperatures, etc. The speed of the engine was maintained at 1800 rpm (60 cps), the intake air temperature was 105 F, and the barometric pressure was 29.2 inches Hg. The specific fuel consumption at 20 kw, when converted to pounds per bhp-hr and corrected to sea-level conditions, compares very favorably with the published performance curve shown in Figure 59.

Figure 63 is a plot of exhaust gas temperatures as a function of load. A range of temperatures is shown rather than a single curve to illustrate the influence of variations in intake-air temperature, barometric pressure, jacket-water and lubricating-oil temperature, speed- and load-setting inaccuracies, etc. The exhaust gas temperature increased with an increase in intake-air temperature with a 10 F increase in intake temperature causing approximately a 20 F increase in exhaust temperature.

Figure 64 shows the rapidity with which exhaust temperatures reach equilibrium with a change in load setting and gives an indication of thermal shocks that must be withstood by the exhaust system. The exhaust system for these tests consisted of 25 ft of flexible 2-1/2-in.-diameter tubing and a muffler. The muffler was located outside the shelter. The total exhaust system pressure loss was 15 in. of water, two-thirds of which occurred in the muffler.

Cooling System Performance

Figure 65 shows the actual amount of heat removed by the jacket-water cooling systems compared with the heat equivalent of the engine shaft power. The scatter of these data reflects the influence of variations in combustion-air inlet temperatures, jacket-water and lubricating-oil temperatures, etc. These data show that all four of the cooling systems tested were capable of satisfactorily cooling the engine under conditions of adequate engine room ventilation and adequate make-up water.

Heat-exchanger effectiveness was calculated for the air-cooled radiator and for the water-to-water heat exchanger using experimental data. The heat-exchanger effectiveness is defined as the ratio of the actual heat transferred to the maximum rate theoretically possible. The effectiveness of the air-cooled radiator was calculated as 25 per cent and that of the water-to-water heat exchanger as 85 per cent. The significance of these values is readily realized when the physical size of an air-cooled radiator is compared with that of a water-cooled heat exchanger for the same engine. For the 20-kw experimental unit, the radiator was about 20 in. wide, 24 in. high, and 2-1/2 in. deep while the water-cooled heat exchanger was only 3 in. in diameter and 16 in. long.

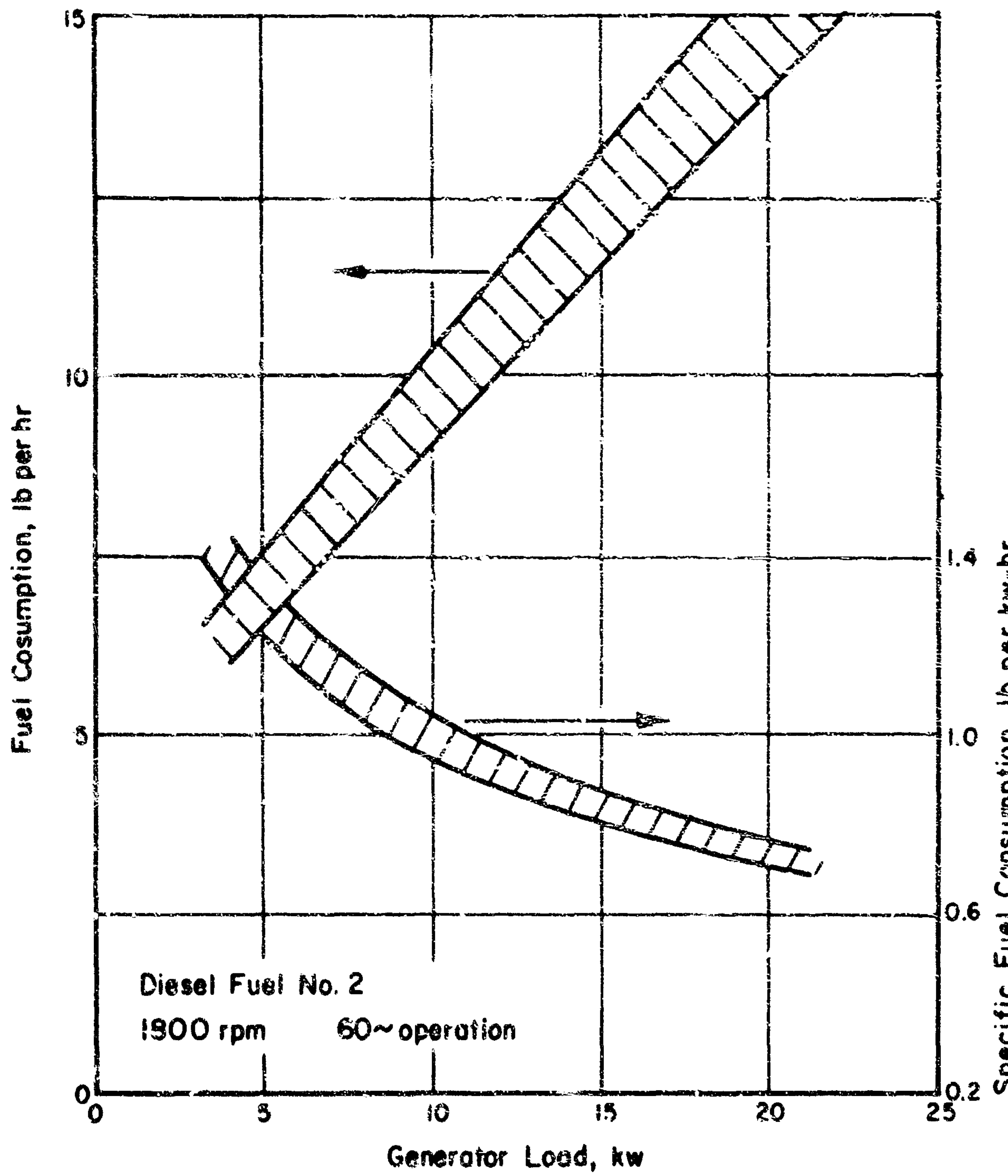

FIGURE 62. FUEL CONSUMPTION

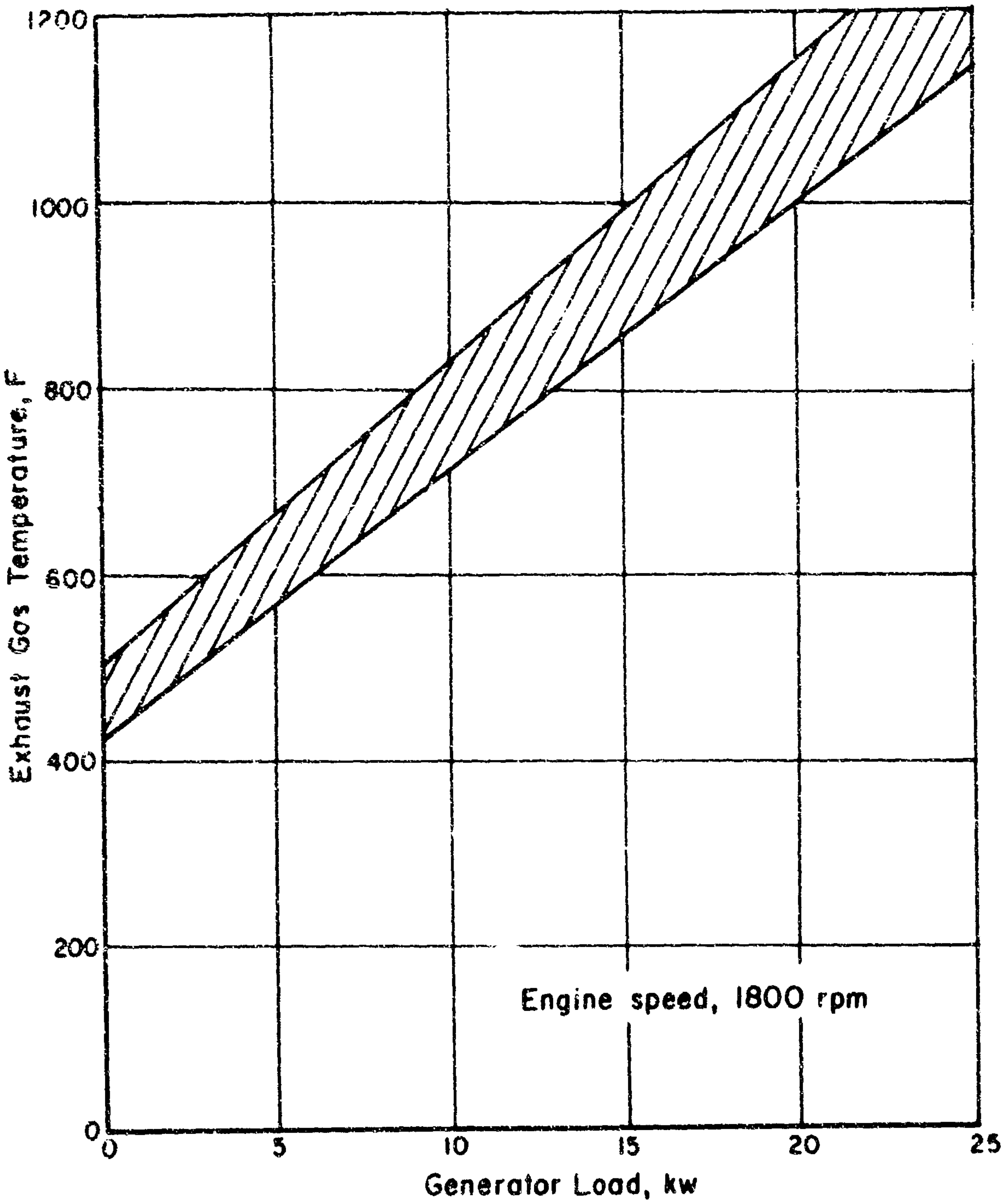

FIGURE 63. EXHAUST GAS TEMPERATURE

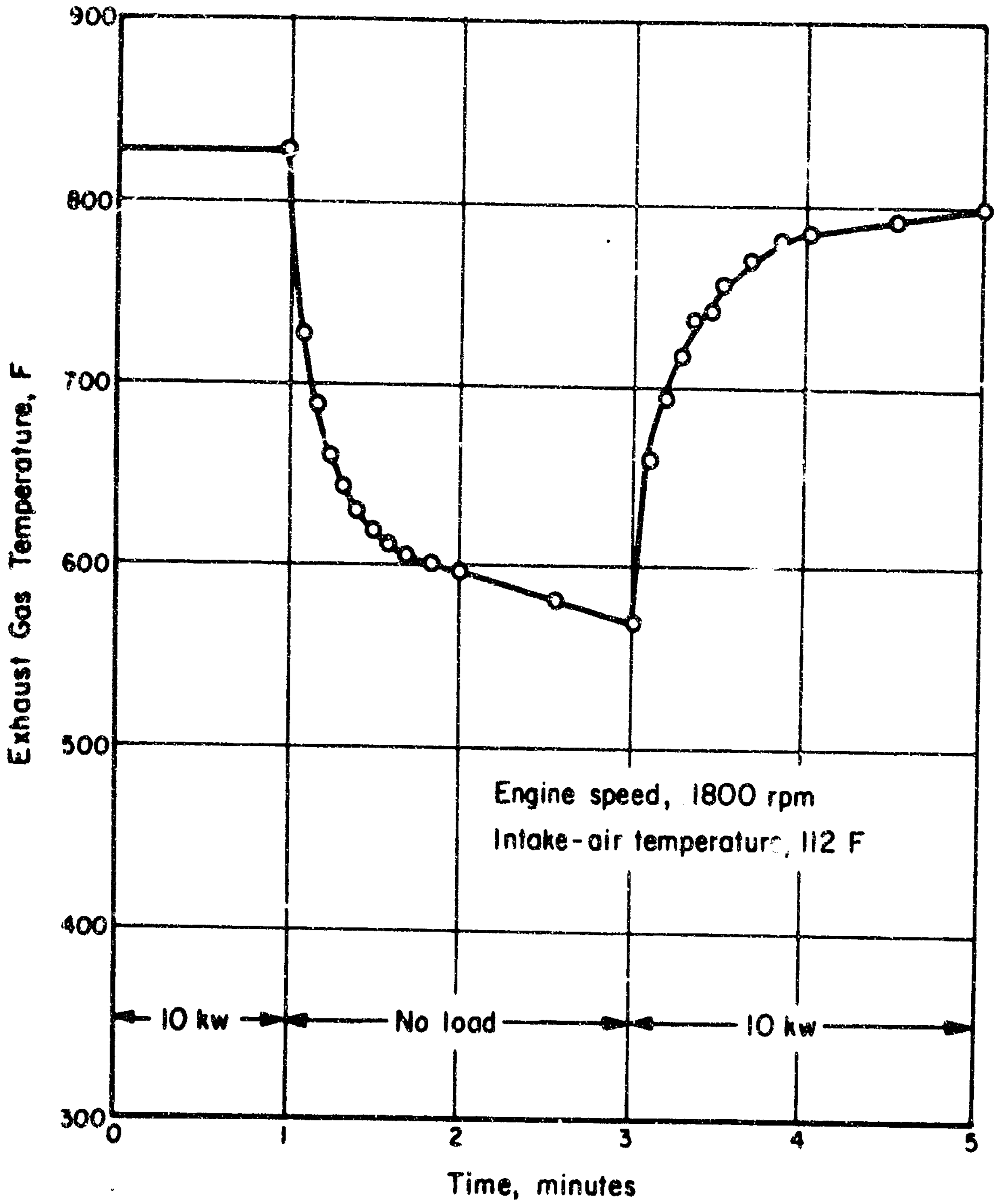

FIGURE 64. EXHAUST-GAS-TEMPERATURE TRANSIENT RESPONSE

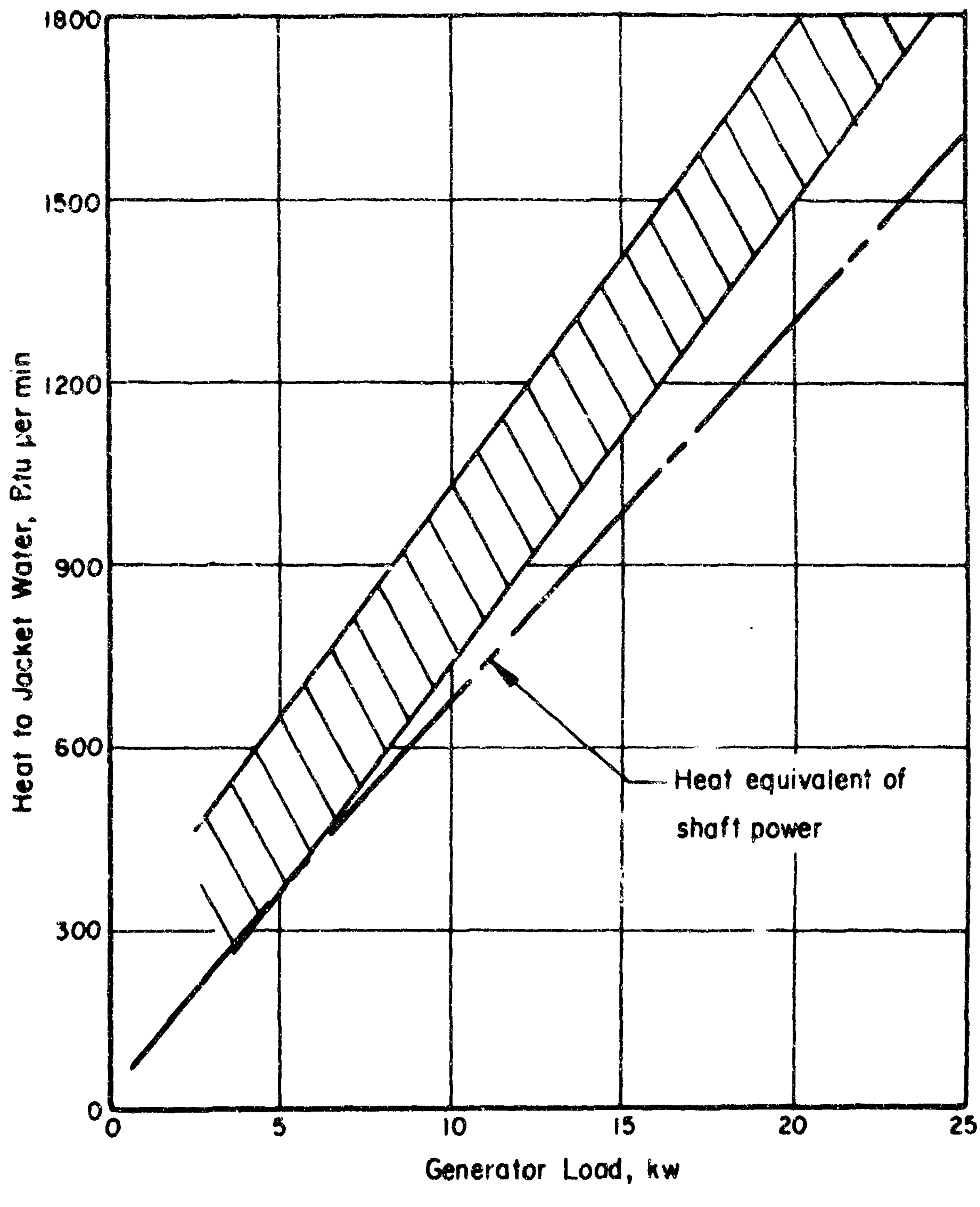

FIGURE 65. COOLING SYSTEMS PERFORMANCE

During the ebullient cooling system studies the water inlet temperature at the engine pump was fairly constant at 205 F and the outlet temperature at 220 F. The separator tank was 12 ft above the engine; hence, the superheated water did not boil until it reached a height in the discharge line where the saturation pressure corresponded to a boiling temperature of 220 F. This occurred at a height of about 6 ft above the engine. Therefore, to prevent boiling within an engine having a water-discharge temperature of 220 F, the steam separator should be located at least 6 ft above the engine if the reservoir is at atmospheric pressure. A slightly pressurized discharge line could be used if it is impractical to achieve vertical height, particularly if it is necessary to mount the steam separator within the engine room. A partially closed valve or orifice in the outlet line would accomplish the same effect; however, a booster pump may be necessary under these conditions to assist the engine water pump in creating the pressure and flow conditions.

The thermostat was removed from the discharge line for these tests as it would have been a restriction. The engine warmed up quite rapidly when an ebullient cooling system was used. During the ebullient cooling system tests the water flow rate was fairly constant at 13 gpm, and the oil temperature did not exceed 240 F. The make-up water rate was 11.9 gal per hr for 20-kw operation, and the heat loss amounted to 1200 Btu per min.

Ventilation System Effects

A series of tests was performed using various quantities of ventilating air to determine the effect of engine room ventilation on engine-generator set performance. In these tests the radiator was mounted on the engine. For one test in this series the engine room was completely closed off and no ventilating air supplied. Following this test, a series of tests were conducted in which the engine room ventilating air was controlled to provide an ambient temperature range of from 80 to 145 F with unrestricted air flow through the engine radiator. A final test was conducted with the radiator air discharged through a duct to the outside. For this test the air flow through the radiator was only about half its normal unrestricted value.

Figure 66 shows how rapidly temperatures rise with an engine operating in an enclosed space with no outside ventilation air supply. The engine fan continually recirculated engine room air through the engine-mounted radiator. No thermostat was used in the engine for this test. After the engine was warmed up to the temperatures shown at zero time, a load of 15 kw was applied to the generator. Within only 14 minutes the room air temperature reached 142 F, the jacket-water outlet temperature was 215 F, and the oil temperature had reached 200 F. The load was removed when the water temperature reached 215 F. Otherwise the water-temperature cutoff switch would have stopped the engine. With normal ventilation, oil temperature generally runs 20 to 25 F higher than water temperature. The curves give no indication that equilibrium conditions were reached at the time of load removal.

For comparison, a similar test was conducted with no ventilation and a load of only 6 kw imposed. However, a 180 F thermostat had been installed, so the water temperature was initially up to 160 F. The load was disconnected after 100 minutes of running, at which time the room air temperature was 150 F, the water-outlet temperature was 200 F, and the oil-sump temperature was 225 F. The test

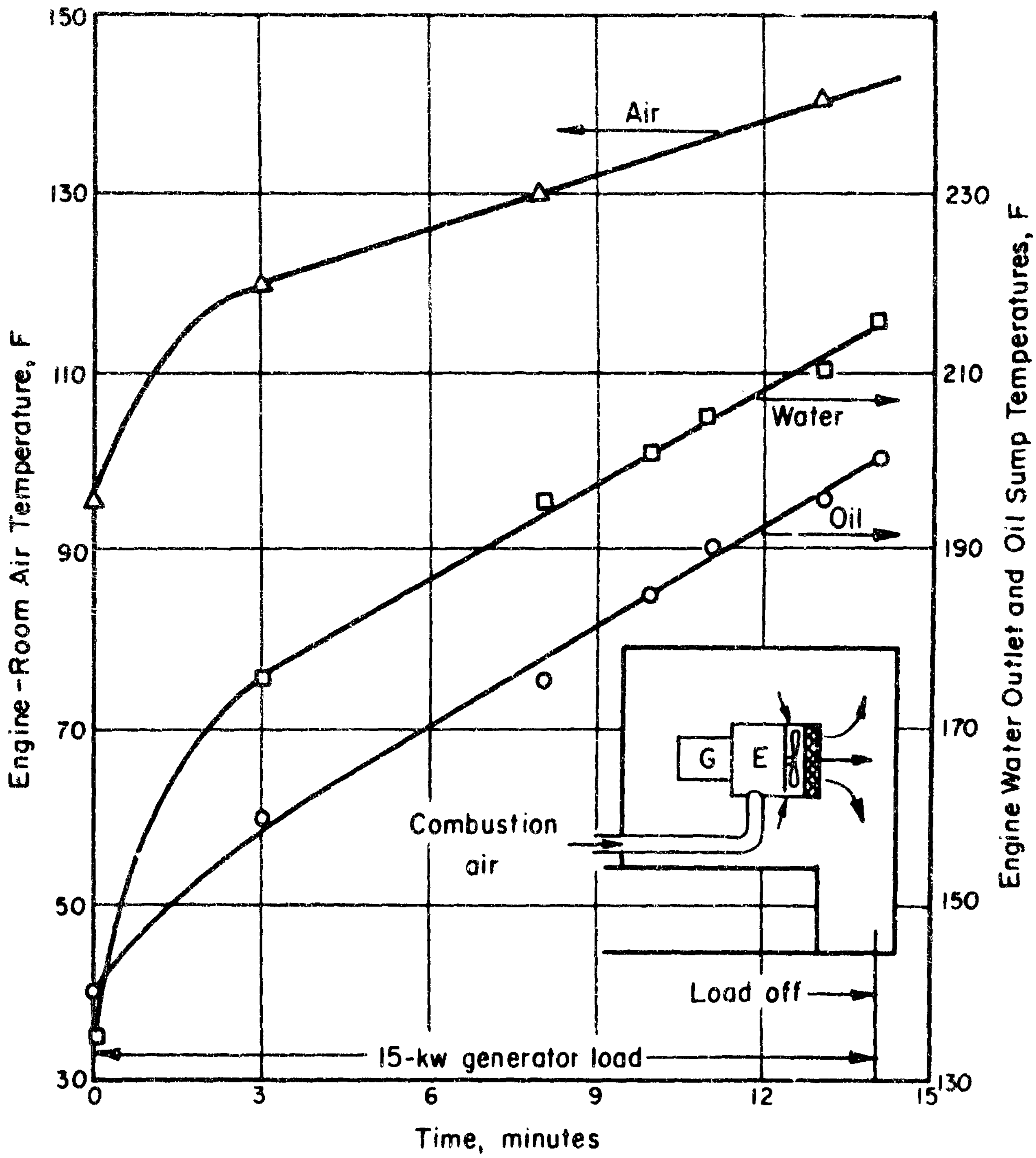

FIGURE 66. TEMPERATURE-TIME VARIATIONS WITH NO VENTILATION

was concluded because of the higher room temperature. Another similar test was conducted using approximately 1900 cfm of outside ventilation air at 62 F and a load of 20 kw. This test was stopped after 36 minutes when the water temperature again reached 215 F. The oil-sump temperature was then 222 F and the ventilation air-discharge temperature was 121 F.

Figure 67 shows the effect of engine room temperature on maximum power output of the demonstration unit. For these tests the engine room ventilation air flow was provided by a separate blower and the engine radiator air flow was unrestricted. For a given boiling temperature limitation, depending on the radiator pressure cap, maximum values of generator load and engine room air temperature were established. Under these conditions full-load output (20 kw) was obtained with 195 F jacket-water temperature and 95 F engine room temperature.

When the air flow through the radiator was reduced to 1800 cfm from the normal flow rate of 3600 cfm, the engine-generator set was able to develop only about two-thirds of full load, at 90 F engine room temperature and 195 F jacket-water outlet temperature. The restriction to radiator air flow was provided by 20 ft of 12-in.-diameter ducting.

Exhaust System Effects

Two test runs at full load were conducted to determine the effect of insulating the exhaust system on the distribution of heat losses from the engine. The radiator was mounted outside for these tests to reduce the room ventilation air requirements. The first test run, lasting about 51 hours, was conducted with no insulation on the exhaust manifold and tubing (flexible tubing about 8 ft in length inside the room). The second test run, lasting about 42 hours, was conducted with a 1 inch thickness of high-temperature insulation covering the exhaust manifold and tubing.

Table 12 shows the approximate heat balances resulting from these tests. A comparison of the data shows that insulating the exhaust system, a relatively simple and inexpensive operation, can provide useful reduction in heat losses to the jacket-water cooling system, to the ventilating air, and to the walls which would be ultimately reflected by reduced heat-sink requirements.

The data presented in Table 12 cannot be considered steady-state data because of the short duration of the tests relative to the anticipated two-week occupancy period.

Noise and Vibration

The sound and vibration measurements were made with the demonstration unit operating in the underground test facility at Battelle. Several means for reducing the noise level in the simulated "occupied" space were experimentally tested and evaluated. Although not originally designed and constructed as a simulated protective shelter, the underground test facility did provide an environment suitable for the noise and vibration tests.

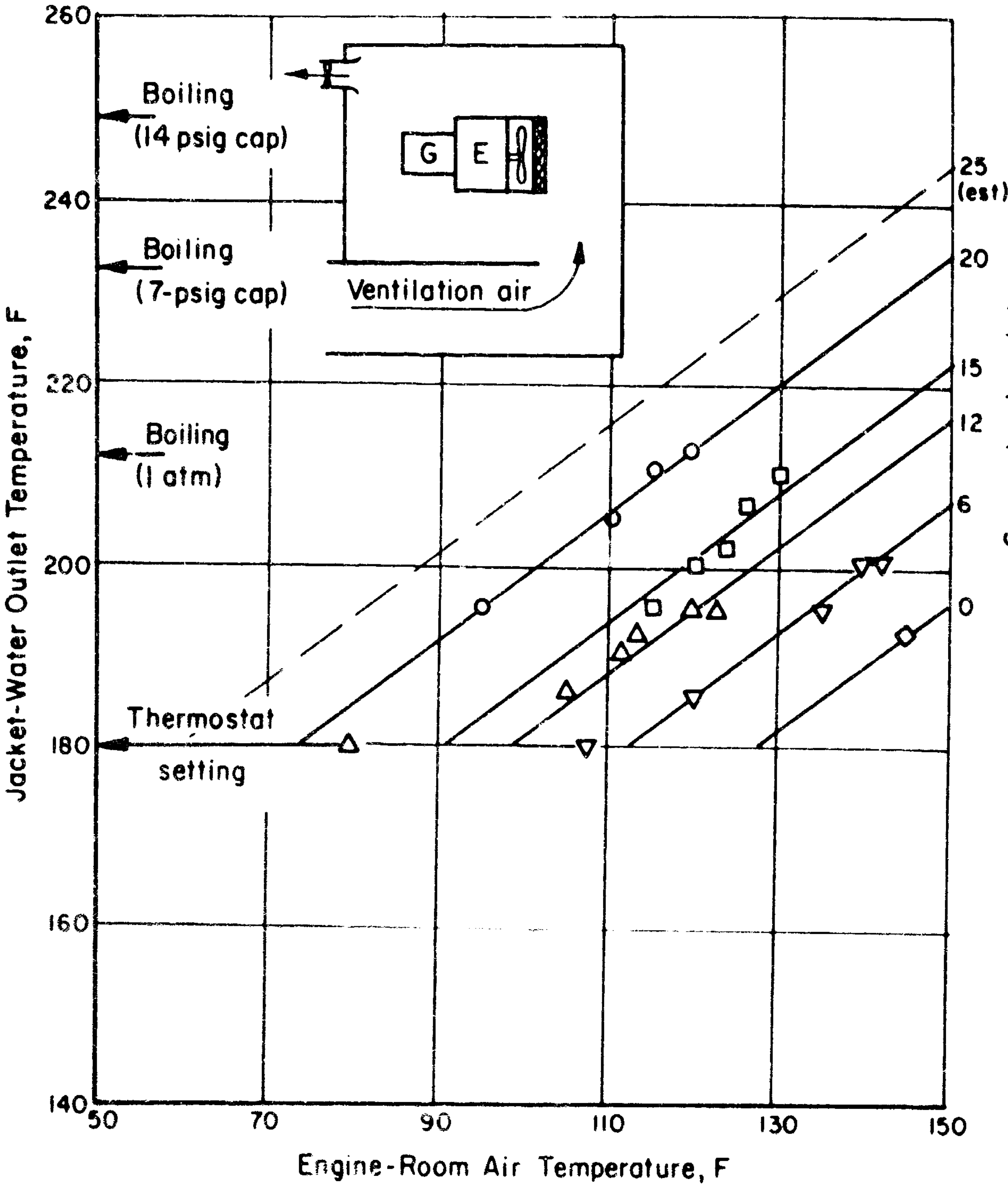

FIGURE 67. EFFECT OF ENGINE-ROOM AIR TEMPERATURE ON MAXIMUM OUTPUT OF DEMONSTRATION UNIT

TABLE 12. APPROXIMATE HEAT BALANCES FOR FULL-LOAD EXHAUST-SYSTEM INSULATION TESTS

Quantity	Uninsulated Exhaust System	Insulated Exhaust System
Heat in fuel, Btu per min	4,580	4,575
Heat to load, per cent	25	25
Heat to outside radiator, per cent	35	34
Heat to exhaust, per cent	21	24
Heat to ventilating air, per cent	10	9
Heat to walls, per cent	9	8
Total, per cent	100	100

In Figure 60 a cutaway drawing of the Battelle underground test facility and the 20-kw demonstration unit was shown. Also shown in the drawing were the aboveground radiator cooling system, the engine room ventilation blower and exhaust duct, a "sound trap" ventilation air inlet duct, the engine room door, and the simulated "occupied" space.

Figure 68 shows the noise criteria curves presented in Figure 57 with experimental data from the demonstration unit sound measurements superimposed. From the plot of sound measurement data which was made in the occupied space of the underground facility with the engine running at rated load and speed, with no sound-absorbing treatment, it can be seen that the noise level was far above that which would be tolerated by humans under normal circumstances. The NCA rating of a given noise is established by the number of the lowest curve tangent to the top of the measured noise spectrum curve. Thus, for the untreated underground facility, the engine produced an NCA rating of nearly 75. If this rating is compared with the recommended rating of NCA 55, it can be seen that this noise would not be acceptable for continuous occupancy.

A small-scale program was initiated to reduce the noise in the occupied space of the shelter. The first need was for a closed door between the engine room and the occupied space. Air flow between the two areas was necessary for engine room ventilation, so a commercial "sound trap" duct was installed in the transom over the door. A refrigerator-type rubber door seal was installed to seal all four edges of the door. All cracks between the sound trap and the walls and door frame were caulked so that the areas were essentially pressure isolated except for the sound trap. The "sound trap" duct was lined with sound-absorbing materials and had a relatively small cross-sectional open area. Its noise reduction effectiveness was about the same as that of the door, a solid-wooden-core, flush-type, 1-3/4-inch-thick outside house door.

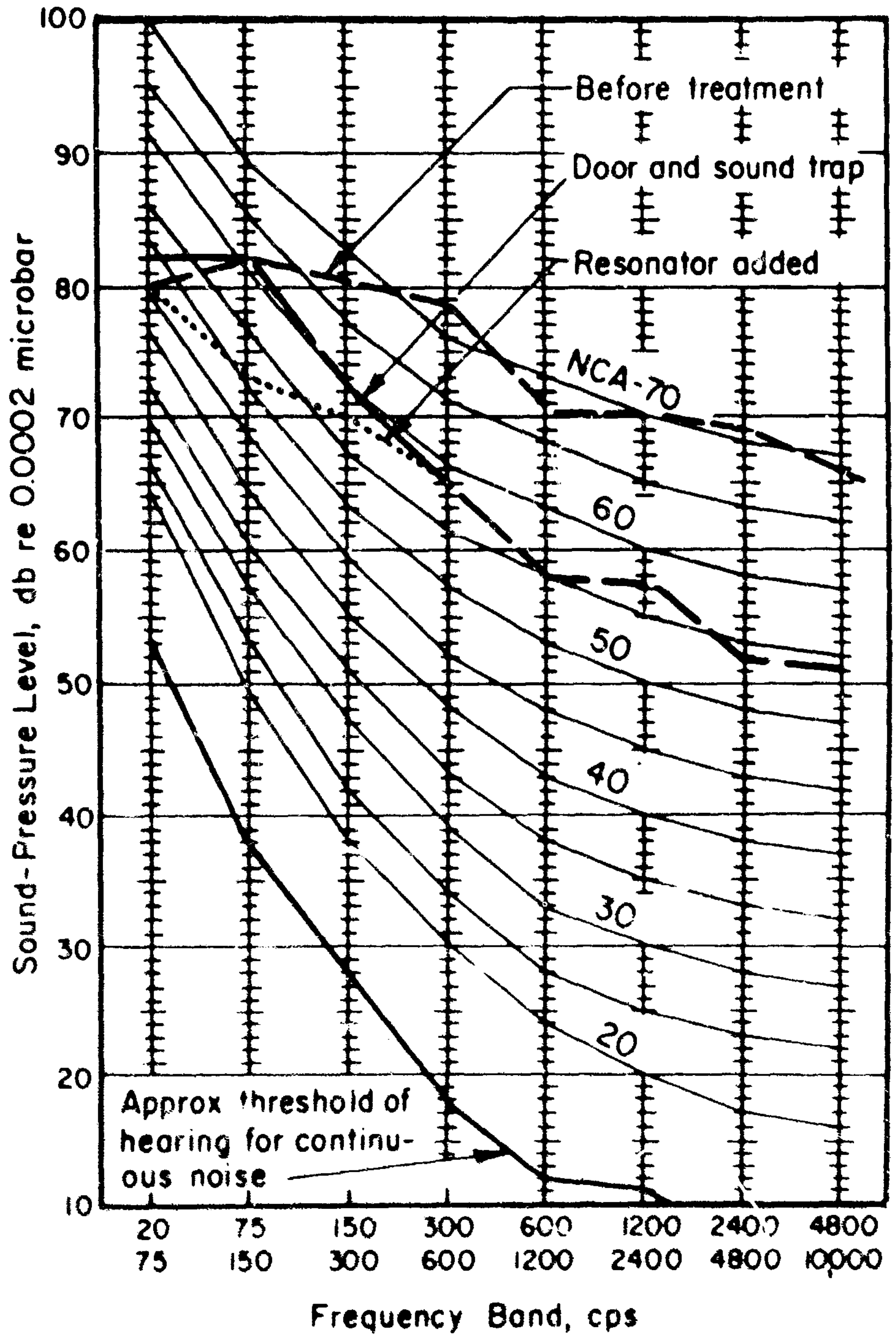

FIGURE 68. ALTERNATE NOISE CRITERIA (NCA) CURVES SHOWING NOISE REDUCTION IN DEMONSTRATION SHELTER

These provisions brought all but the low frequencies down below the NCA 60 curve, as is shown by the solid line in Figure 68, and greatly reduced the harshness of the noise. To reduce the fairly high noise peak at 120 cps, a simple Helmholtz resonator was constructed from a 20-gal container and a short section of

duct joined to the engine room end of the sound trap duct. After the resonator volume was adjusted for maximum cancellation of the 120-cps peak by adding water to the container, the full noise spectrum in the occupied space was below the NCA 60 curve, as shown in Figure 68 by the dotted line.

Although this final sound spectrum exceeded the recommended NCA 55 rating, it is believed that the type of noise-reduction treatment used in the Battelle facility would be adequate for fallout shelters. This same amount of sound power represented by an NCA rating of 60 in the simulated occupied space would probably yield an NCA rating of 55 or less in any reasonable sized shelter living area because of the greater sound absorption in larger occupied enclosures.

Vibration measurements indicated that in an underground concrete enclosure vibration from the prime mover will not be directly annoying to people. The vibration levels were well below annoyance levels even with the engine-generator skid resting solidly on the floor. Anchor bolts were not used with either solid mounting or vibration-pad mounting. However, no movement of the skid relative to the floor was observed. When vibration pads were used the set vibrated much more than when it was directly in contact with the floor. This vibration could cause a reliability problem with the engine, generator, or other components. Peak accelerations up to 6 G's were measured on the control box of the engine-generator set and an electric wire broke after only 50 hours of running. This type of vibration problem is related more to the general quality of manufacture than to any precautions the shelter designer could take.

Other Test Results

Maximum power tests performed in the laboratory revealed that the maximum power capability of the engine as delivered was 40 corrected bhp at 1800 rpm. This power output is equivalent to the intermittent power rating of the engine as shown in Figure 59. The maximum laboratory performance power rating of 45 corrected bph at 1800 rpm, also shown in Figure 59, could not be obtained in our tests. Therefore, it is assumed that the manufacturer specifically prevents excessive power operation by limiting the fuel pump and governor control settings. Internal readjustment of the fuel pump and governor would probably result in the 45 hp capability.

Kerosene was tested in the engine as an alternative fuel. The engine ran very well using kerosene but was unable to deliver the same maximum power as with No. 2 diesel oil. Although the heating value of kerosene is slightly higher on a weight basis than No. 2 diesel oil, on a volume basis it is slightly lower. Therefore, with a constant displacement fuel pump the total heat added to the engine as fuel is lower using kerosene. The maximum shaft power attained using kerosene was 39.0 hp corrected to standard conditions. This represents a reduction in maximum horsepower output of 3 per cent.

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